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THE MEETING OF THE FLEETS.

By E. H. B. Anderson, Associate.

The third anniversary of the entry of the United States of America into the great European war has passed.

The first event of importance in the interests of the Allies was the arrival of the Eighth Division, Destroyer Force, Atlantic Fleet, in British waters on the morning of May 4th, 1917.

These vessels were the fore-runners of the large American fleet which helped to bring victory on land and sea.

The flagship was U. S. S. *Wadsworth*, No. 60, in charge of Commander Joseph K. Taussig, who was Senior Officer.

The other vessels in the Division and their commanding officers were as follows:

Lieutenant Commander Ward K. Wortman, U. S. S. *Porter*, No. 58.

Lieutenant Commander Rufus F. Zogbaum, U. S. S. *Davis*, No. 65.

Commander Alfred W. Johnson, U. S. S. *Conyngham*, No. 59.

Lieutenant Commander Arthur P. Fairfield, U. S. S. *McDougal*, No. 54.

Lieutenant Commander Fred H. Poteet, U. S. S. *Wainwright*, No. 62.

The Division was met off Queenstown, Ireland, by the British destroyer *Mary Rose*, having on board Captain E. R. G. R. Evans, R. N., representing the Admiralty, and who, a few days prior to the arrival of these destroyers, had distinguished himself when commanding H. M. S. *Broke*.

The picture representing the arrival, "The Meeting of the Fleets," was made by Charles Pears, Esq., a well-known English artist, and this illustrates in a striking manner a great historical event for the records of the Navy.

On arriving at Queenstown, the Lord Mayor of Cork and many of the leading citizens welcomed Commander Taussig and his brother officers.

On the conclusion of this ceremony the officers visited Admiralty House and reported for duty to Vice-Admiral Sir Lewis Bayly, under whose command they carried out the various duties assigned to them for over one year and a half.

A remarkable feature in regard to the arrival of these vessels was, that they were ready to proceed to sea immediately on completing the work of refilling the oil fuel tanks. However, four days were allowed for overhauling machinery, taking on stores, etc., and then the six vessels proceeded to sea for the regular duty of six days out and two days of rest between trips.

These vessels represented the latest practice of the destroyer builders' products: three of them coming from the shipyard of Bath Iron Works, Ltd., Bath, Maine; two from The William Cramp & Sons Ship and Engine Building Company, Philadelphia, Pa., and one from The New York Shipbuilding Corporation, Camden, N. J.

They were capable of steaming 30 knots, all were driven by Parsons Turbines, operating twin screw propeller wheels. The propelling machinery of *Wadsworth* consists of twin sets of geared turbines, the gearing having the unusual feature of a single helix, instead of the standard right and left hand herring-bone type. The other five have the go-ahead turbines arranged in series, a high-pressure turbine driving the starboard shaft and a low-pressure turbine driving the port shaft; this arrangement of machinery improves the economy of the propelling machinery considerably.

It is not too much to say that the *Wadsworth* has shown a higher economy of fuel consumption per knot run than probably any other destroyer ever built, and the novel feature of the gearing has fully justified all that the designers claimed for this design.

The dimensions of the vessels are almost alike:

Length on L. W. L., feet, 310.

Breadth on L. W. L., feet, 29.75.

Draught on L. W. L., feet, 9.33.

Displacement, corresponding, tons, 1,075.

The oil fuel tanks have a total capacity of 300 tons, and the machinery was designed to give a cruising radius of 2,500 knots at 20 knots speed.

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U. S. S. *Davis*, completed in 1916.

INVESTIGATION OF WATER-AIR RADIATORS FOR COOLING GENERATORS AND MOTORS.*

By H. G. Reist,

Alternating-Current Engineering Department, General Electric Co.

and

E. H. Freiburghouse,

Turbo-Alternator Engineering Department, General Electric Co.

The satisfactory solution to the problem of ventilating large capacity electrical units in land stations, by means of ducts, air washers, etc., is not as readily applicable to the cooling of electrical apparatus where extreme compactness is essential—as for instance in the propelling machinery of a ship. The use of the customarily large ventilation air ducts would tend to offset one of the great advantages of electric marine drive; viz., the enlargement of the cargo carrying space. The following article reviews an investigation made to overcome the difficulty by applying a closed system of air ventilation with water-air radiators for removing the heat from the circulating system.—Editor.

SPECIAL CONDITIONS OF VENTILATION.

Under favorable conditions the common arrangement of ducts and air washers, which permits of a complete control of the ventilation for turbo-alternators and stations, is believed to be best. However, cases sometimes exist where it is very desirable to eliminate the numerous long air ducts of large cross-section. This is especially apparent in the case of electrically-driven vessels where lack of space and other reasons make ducts objectionable. In order to meet such special conditions, it is essential that a closed system be employed for the circulation of the cooling air.

* Reprinted from "General Electric Review," February, 1920, with the permission of that publication.

HEAT OF LOSSES MUST BE REMOVED FROM THE AIR.

For seagoing vessels, a closed system of ventilation is desirable. However, with such a system it becomes necessary to cool the circulating air through the same range of temperature that it is raised in passing through the generator or motor it cools.

The quantity of air which passes through a turbo-alternator is so related to the electrical and windage losses of the machine that the rise in temperature of the cooling air, or the range through which it must be cooled if used again, is approximately 20 degrees C. Assuming the standard temperature of the ingoing air to be 40 degrees C., as established by the Rules' Committee of the A. I. E. E., the temperature of the air which leaves the generator is 60 degrees C., and 50 degrees C. is the mean of the ingoing and outgoing air of any cooling device in the ventilating circuit.

COOLING METHODS.

The apparatus used for the removal of heat from the air will depend somewhat upon the character of the cooling water. Under most conditions, however, with a plentiful supply of fresh water, the air can readily be cooled by the use of the spray form of air washer now so generally employed for the cleaning and cooling of air. The cooling water in this case is discarded or, if necessary, recooled and used again. As the spray washer is an apparatus in which the air comes in contact with finely divided water particles, formed by spray nozzles, the character of the water supplied will determine whether an air washer would be applicable.

On ocean vessels, sea water is the only available heat conveyor from the cooling device of a closed system of ventilation. It is quite doubtful whether it is safe to have the cooling air pass continually through a spray of salt water, on account of the liability of small quantities of salt being carried into the windings.

In such cases two courses are open: either to recirculate a relatively small quantity of fresh water through the air washer and cool this water outside of the washer through the intermediation of a cooler, or to install some form of radiator in the ventilating air circuit and use salt water directly in the radiator tubes. This latter scheme was considered by one of the writers nearly one year ago as a feasible solution of a ventilating problem where it was very difficult to provide external ducts of sufficient size, and where an air washer was impossible because of the character of the cooling water. An investigation was started which led to tests from which the results given in this paper were obtained.

The function of the automobile radiator is to cool the circulating water and to heat the air passing through the radiator core or, in other words, to transfer heat from the water to the air. For the purpose under consideration, this heat transfer process would be reversed since the heat is to be transferred from the cooling air of the generator to the water in the tubes.

SELECTION OF THE RADIATOR.

In those fields of service where radiators are given hard usage, the fin-and-tube type seems to be almost universally used. Clogging of irregular water channels by foreign matter is the most common cause of trouble in radiators. Obviously, the round tube offers the least trouble from clogging and, for given weight, is of the strongest construction as a conductor of water.

It was for these reasons that a small fin-and-tube type radiator was selected, and tests made to determine the rate of heat transfer as a function of the rates of flow of water and air.

TEST OF RADIATORS.

The tests conducted upon this small radiator having one-quarter inch tubes confirmed the opinion that the scheme was feasible; in fact, they indicated results better than were ex-

pected. The reduction in the diameter of radiator tubes has a double effect in accelerating the transmission of heat, since it not only reduces the travel of the water particles between successive contacts with the tube wall, but also increases the area of contact surface of a unit weight of the water. Although it is beneficial to use tubing of small diameter for these reasons, it was nevertheless considered advisable to use tubing of larger diameter than one-fourth inch in order to reduce the liability of clogging. This justified a more extensive series of tests which were made upon radiators having tubes of greater diameter and fins of different design. These tests were made to determine the relative dimensions of radiators, the heat transfer as functions of the speed of the air and the water, the amount of cooling surface for a given duty, and the resistance of the radiator to the flow of the air and the water.

Since the frontal area of a radiator within a restricted space would be limited, thus determining the minimum speed of the air through the core, it was realized that the practical application of radiators for this purpose greatly depended upon whether the air could be forced through the necessary depth of core without prohibitive pressure.

Five large radiators, intended for truck service and fitted with special headers, were obtained through the courtesy of the G. & O. Manufacturing Co., of New Haven, Conn. Fig. 1 shows one of these sections as described in Table I. The

TABLE I.

Tanks were constructed of No. 18 B. & S. copper.	
Shaped 4 15/16 by 3 by 23½ inches.	
Core had six rows of 29 tubes, tubes	174
Diameter of tube (outer), inch	0.375
Diameter of tube (inner), inch	0.339
Length of tube, inches	23 27/32
Length of tubing per section, feet	246
Fins outside, inch	0.75 by 0.75 by 0.007
Spacing of fins, inch pitch	0.165

Spacing of tubes front and back, inch	0.8125
External surface of tube per linear foot, square inches	13.55
Surface of fins (total per linear foot of tubing), square inches	67.3
Total cooling surface per linear foot of tubing, square inches	80.85
Equal to, square foot	0.5615
Free air section (between tubes and fins) of frontal area, square feet	2.02
Frontal area of core, square feet	3.90
Weight of core empty per cubic foot, pounds	37
Weight of core plus water per cubic feet, pounds	45.9

radiators were installed in an air tunnel in series with the air and water flow as indicated by Figs. 2 and 3. An end view of the entire testing equipment used is shown in Fig. 4.

DESIGN OF G. & O. TRUCK RADIATOR.

The header tanks at each end of the core were specially designed for test purposes.

A motor-driven blower of known speed, pressure, and volume characteristics delivered air through electrically-heated grids to the radiators.

AIR FLOW.

The air flow through the radiators was varied from 0.61 to 1.9 pounds per second per square foot of the frontal area of the core, which corresponds to a range in velocity of 930 to 2,880 feet per minute through the minimum free air section of the core.

The quantity or mass flow was determined by means of a pitot tube and inclined manometer, also by the rise in temperature of the air and the electrical input as the air passed through the heating grid.

AIR PRESSURE.

Tests were made to determine the fan pressures required to deliver air through the five-section radiator. Ordinarily a radiator for service could be so designed that the depth need

FIG. 1.—PHOTOGRAPH OF TRUCK RADIATOR FITTED WITH SPECIAL HEADER TANK FOR TEST PURPOSES. FOR DIMENSIONS SEE TABLE I.

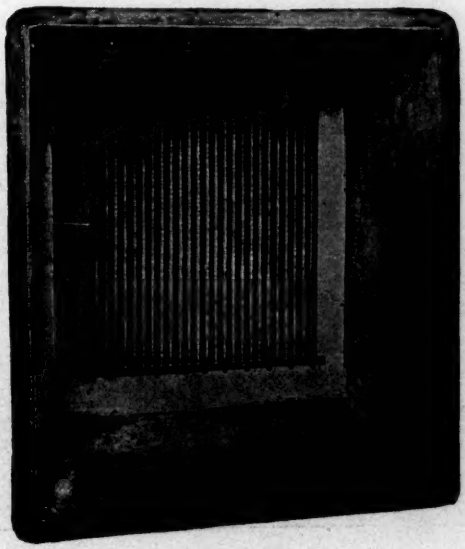


FIG. 2.—ONE OF THE RADIATOR SECTIONS IN THE AIR TUNNEL SURROUNDED BY BAFFLES. BAFFLES AT FRONT AND SIDES OF SECTION NOT SEEN.

FIG. 3.—SIDE VIEW OF AIR TUNNEL AND WATER PIPING OF THE RADIATORS.

FIG. 4.—END VIEW OF THE AIR TUNNEL AND DISCHARGE DUCT.

not exceed that equal to three of the sections tested, also the frontal area could be such as to limit the air velocity through the radiator core to less than 2,000 feet per minute, which would add a resistance in the air circuit equal to about 2.5 inches of water.

If the speed of the fan and the resistance of the air circuit are fixed, the volume of the air and the air pressure developed by the fan are also fixed. Moderate increases in the resistance of the fan circuit do not appreciably affect the pressure developed by the fan, as only part of the fan pressure is used in passing the air through this additional resistance. For moderate changes in resistance the volume of the air may be taken as proportional to the square root of the ratio of the pressure available to pass a given quantity of air through the generator alone under the two conditions. For instance, consider a case where a certain machine has no external duct and the fan produces a pressure of 13 inches of water while passing 60,000 cubic feet of air per minute through the machine. Assuming that an air duct and radiator are added in which the total head lost is three inches, then the quantity of air under the second condition is closely given by

$$Q = \sqrt{\frac{13-3}{13}} \times 60{,}000 = 52{,}700$$

cubic feet per minute.

Since long air ducts having numerous bends or changes in cross-section are eliminated by the use of the radiator and short ducts, it will be possible in many cases to obtain the necessary quantity of air without the use of external blowers.

The relation of air pressure to air velocity through the radiator cores is indicated by Fig. 5.

Beyond a certain air velocity the loss in air pressure required to force the air through the radiator makes the gain in heat transfer prohibitive.

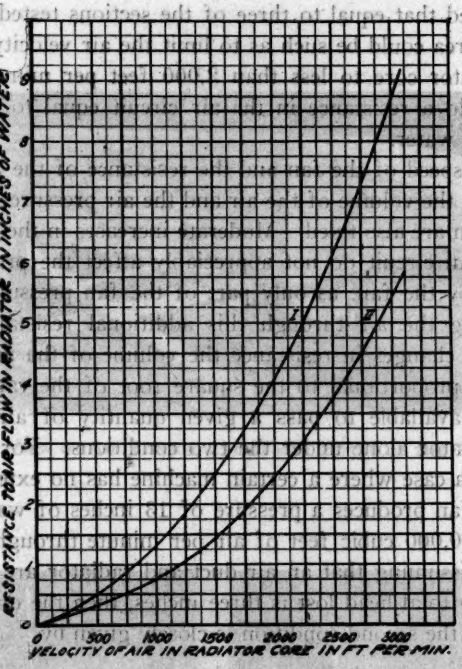


FIG. 5.—IMPACT PRESSURE IN INCHES OF WATER TO FORCE AIR THROUGH THE RADIATORS AS A FUNCTION OF THE VELOCITY OF THE AIR IN THE FREE AIR SECTION OF THE CORE.

CURVE I REQUIRED BY FIVE SECTIONS IN SERIES.
CURVE II REQUIRED BY THREE SECTIONS IN SERIES.

TEMPERATURE OF THE AIR.

The temperatures of the air were obtained over the sections of air flow at the front and back of the heating grid and over each of the five sections of the radiator by means of extensive temperature coils and numerous thermometers. These furnished data for determining the mean temperatures of the air for the various sections of radiator.

WATER FLOW.

The flow of water through the five sections of the radiator in series was varied from 8.25 to 40.5 gallons per minute, corresponding to a range in velocity of 10.1 to 49.6 feet per minute in the tubes. Entering at the bottom of the last or fifth section of the core, the water flowed upward in all of the 174 tubes of this section, then alternately downward and upward through the remaining sections, leaving the radiator from the top header tank of the first section. The air flowed through the cores of the five sections from front to back, thus moving through the core in the direction opposite to the progression of the water.

In determining the curves shown in Fig. 6 it is seen that

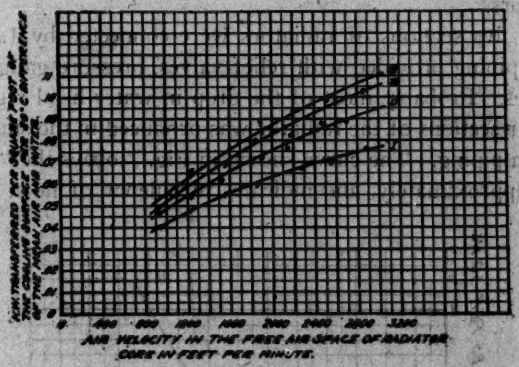


FIG. 6.—RATE OF HEAT TRANSMISSION AS A FUNCTION OF THE SPEED OF AIR THROUGH THE FREE AIR SECTION OF THE RADIATOR CORE.

CURVE I. FOR AN APPROXIMATE SPEED OF WATER IN THE TUBING EQUAL TO 10 FT. PER MINUTE.
CURVE II. FOR 30 FT. PER MINUTE.
CURVE III. FOR 40 FT. PER MINUTE.
CURVE IV. FOR 50 FT. PER MINUTE.

the observed points for the higher flows do not lie as regularly on the curves as do those for the lowest flow. The curves were determined from the temperature rises of the water,

which were lower for the greater quantities of water and thus made a greater percentage error in the observed temperatures. Nevertheless, it is shown that for an increase in the water flow up to a certain rate there is obtained an increase in the heat transfer for a fixed difference of the mean temperatures of the air and water, after which little benefit is secured by a further increase in the velocity of the water in the tubes.

Temperatures were obtained in the water circuit on each side of each of the five sections of the core, which made known the relative utility of the sections throughout the depth as a function of the air speed through the core. The curves shown in Fig. 6 were estimated from data taken on the first three sections of the core, since it was realized that this would be the approximate depth in the direction of air flow on most radiators.

The five sections of radiators were connected by 1.25-inch water piping provided with relief valves, pressure gauges, and U-tubes. Probably most of the drop in water head registered on four sections of the radiator was produced in the small connection piping. The curve shown in Fig. 7 obeys the "square law" approximately, and indicates turbulency of flow.

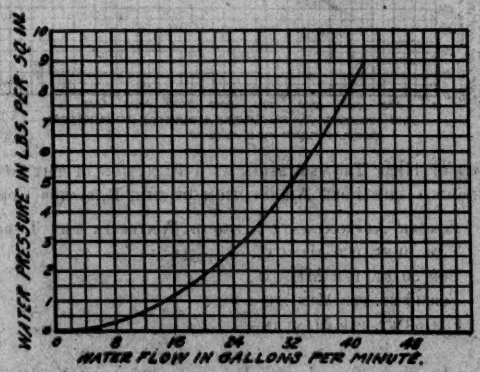


FIG. 7.—DIFFERENCE IN WATER PRESSURE IN POUNDS PER SQUARE INCH OBTAINED ON FOUR SECTIONS OF THE CORE AS A FUNCTION OF THE WATER FLOW.

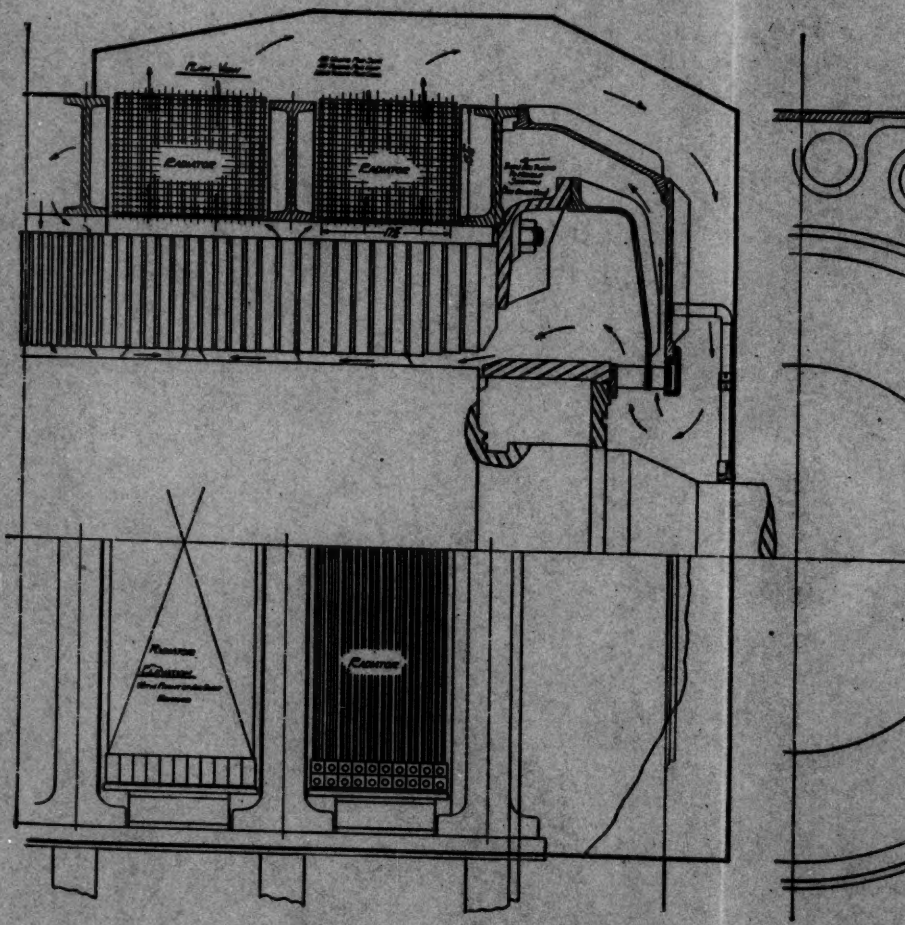
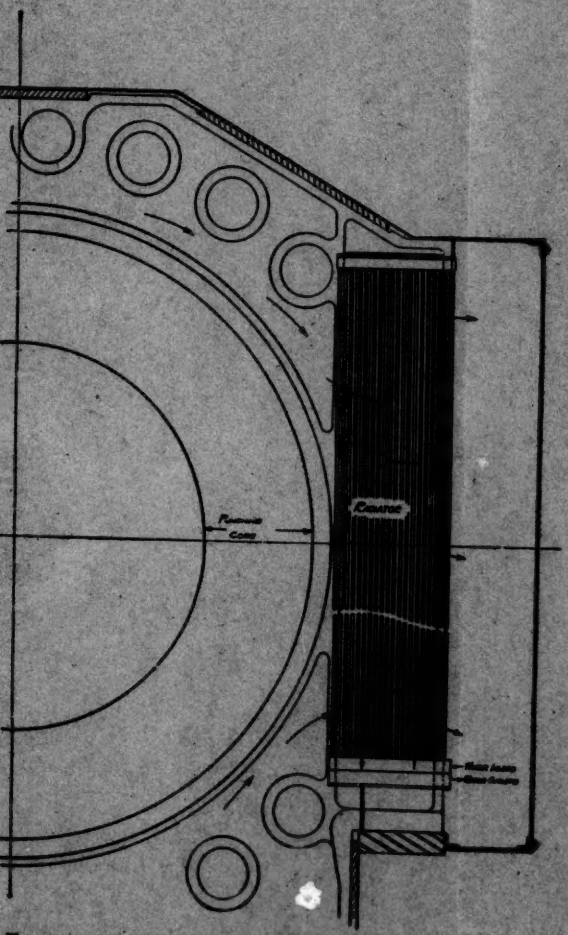


FIG. 8.—SECTIONALIZED RADIATOR ARRANGED WITHIN THE FRAME OF A



FRAME OF A TURBO-ALTERNATOR.

ARRANGEMENT OF RADIATORS WITH GENERATOR.

Figs. 8 and 9 indicate possible arrangements of radiators between the sections of the stator frame, or entirely outside

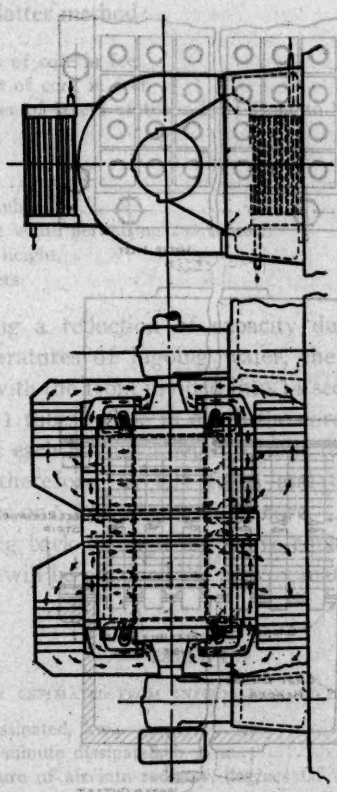


FIG. 9.—INDICATES THE VENTILATION SYSTEM OBTAINED WITH A TURBO-ALTERNATOR AND A RADIATOR FOR COOLING THE AIR.

of it with the necessary inlet and discharge ducts for the air. As indicated by these diagrams, the radiator would be divided into a number of sections or units, thus facilitating handling or repairing a section while the remainder of the radiator was in service.

Fig. 10 indicates an arrangement of the tubes in a flexible tube sheet, to which is secured the header tanks. The flexible tube sheet has been found most satisfactory to prevent leaking

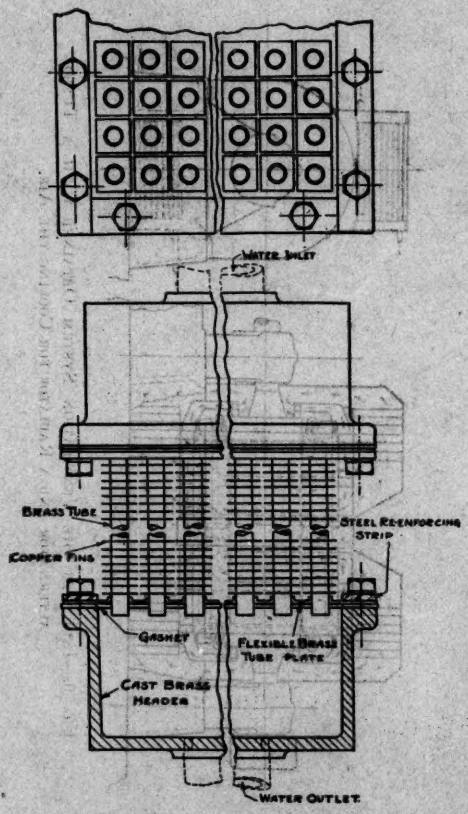


FIG. 10.—DETAILS OF A RADIATOR SECTION.

tubes. Radiators intended for use with sea water should be constructed of the metal which will best resist the corrosive action.

In order to agree with the values of Table II, the width, height, and number of layers of tubes must be found by trial or by solving through simultaneous equations relating the unknowns.

Using the latter method:

Let x = width of core in feet.
Let y = height of core in feet.
Let z = number of layers of tubes from front to back of core.

Equations for:

Frontal area ..... $47.8 = 0.518\,x\,y$
Surface ..... $9,770 = 83\,x\,y\,z$
Number of tubes ..... $3,602 = 14.75\,x\,z$

$x = 19.15$ feet width permitting 283 tubes.
$y = 4.82$ feet height.
$z = 12.75$ layers.

Anticipating a reduction of capacity due to clogging or higher temperatures of ingoing water, the radiator may be constructed with the front divided into 14 sections each having a width of 21 tubes, while in depth the core may be made in two divisions each having a depth of nine tubes. The frontal width will, therefore, be 294 tubes and the depth will be eighteen tubes or 5,292 tubes total in core.

Substituting back through the above equations and computations there will be obtained the approximate relations shown in Table III.

TABLE II.

RADIATOR DESIGN ESTIMATED FROM INFORMATION SECURED FROM THE TESTS.

(1)	Power dissipated, kw	600
(2)	Heat per minute dissipated, b. t. u.	34,200
(3)	Temperature of air into radiator, degrees C.	55
(4)	Temperature of air out of radiator, degrees C.	37
(5)	Temperature of air mean, degrees C.	46
(6)	Temperature of air drop, degrees C.	18
(7)	Temperature of water into radiator, degrees C.	25
(8)	Temperature of water out of radiator, degrees C.	28
(9)	Temperature of water mean, degrees C.	26.5
(10)	Temperature rise of water, degrees C.	3
(11)	Mean air minus mean water, degrees C.	19.5

(12) Velocity of water in tubes, feet per minute	50
(13) Velocity of air in free air area of core, feet per minute	1,200
Cooling Surfaces	
(14) External surface of tube per linear foot, square inches	13.55
(15) Surface of fins per linear foot of tube, square inches	67.3
(16) Total surface per linear foot of tube, square inches	80.85
(17) Total surface per linear foot of tube, square feet	.5615
(18) Total cooling surface per sq. ft. of frontal area in one layer of tubes, square feet	8.3
(19) Total cooling surface required under conditions (12) and (13) is from $S \times 0.0614 = 600$ kw. $S =$ Square ft.	9,770
Estimated Weights, Volumes, and Surfaces	
(20) Weight of air required per minute based upon (6) and (2), pounds per minute	4,310
(21) Volume of air from (20), cubic feet per minute	57,500
(22) Free air area required from (20) and (13), square feet	47.8
(23) Weight of sea water based upon (2) and (10), pounds per minute	6,670
(24) Number of tubes required to carry water based upon bore of tube (0.083 sq. in.), also (12) and (23), tubes	3,602
(25) Spacing tubes 0.8125 in. apart or, per foot	14.75

TABLE III.

RADIATOR DESIGN FOR DIFFERENT CONDITIONS FROM THOSE OF TABLE II.

(1) Power dissipated, kw	600
(2) Temperature of air ingoing, degrees C	50
(3) Temperature of air outgoing, degrees C	32
(4) Temperature of water ingoing, degrees C	25
(5) Temperature of water outgoing, degrees C	27
(6) Quantity of water at 50 ft. per minute, pounds per minute	9,800
(7) Width of core, feet	19.94
(8) Height of core, feet	4.82
(9) Volume of core, cubic feet	117.5
(10) Weight of core empty, pounds	4,340
(11) Weight of core plus sea water, pounds	5,400
(12) Velocity of air in core, feet per minute	1,150
(13) Resistance to air flow in inches of water, inch	1.0

PROPELLERS FOR LOW-POWERED MERCHANT SHIPS.

By Commander S. M. Robinson, U. S. N., Member.

The use of new forms of propulsion has brought up anew the question of propeller design for this type of ship. There are, at present, more different methods of propulsion advocated for this type of ship than for any other. There are the reciprocating steam engine, the Diesel engine, geared turbines, and various forms of electric propulsion. The first of these methods of propulsion will require a very low-speed propeller, the Diesel engine will require a higher-speed propeller, and for the latter two methods the higher the speed of the propeller the more suitable it will be for the machinery.

So many ships of this type have been built that the propeller used has become almost standard; a large diameter, 4-bladed propeller of low revolutions has always been used. It has been customary to use the largest diameter of propeller that could be swung, on the assumption that this would be the most efficient. While this assumption is generally true, and may be taken as always true if the R.P.M. are definitely fixed, it may very well be that the R.P.M. can be varied and in this case there will be conditions where a smaller diameter propeller of higher R.P.M. will have as good an efficiency as a larger diameter propeller of lower R.P.M.

This can best be illustrated by actually solving for various propellers for a vessel of this type. Assume a tanker to have a "slip block coefficient" of .8 and to require 1,500 effective horsepower at 11 knots speed. Assume that a maximum propeller diameter of 18 feet can be used. From Chart 22 B of Dyson's "Screw Propellers" we find that the values of $\frac{\text{e.h.p.}}{\text{E.H.P.}}$

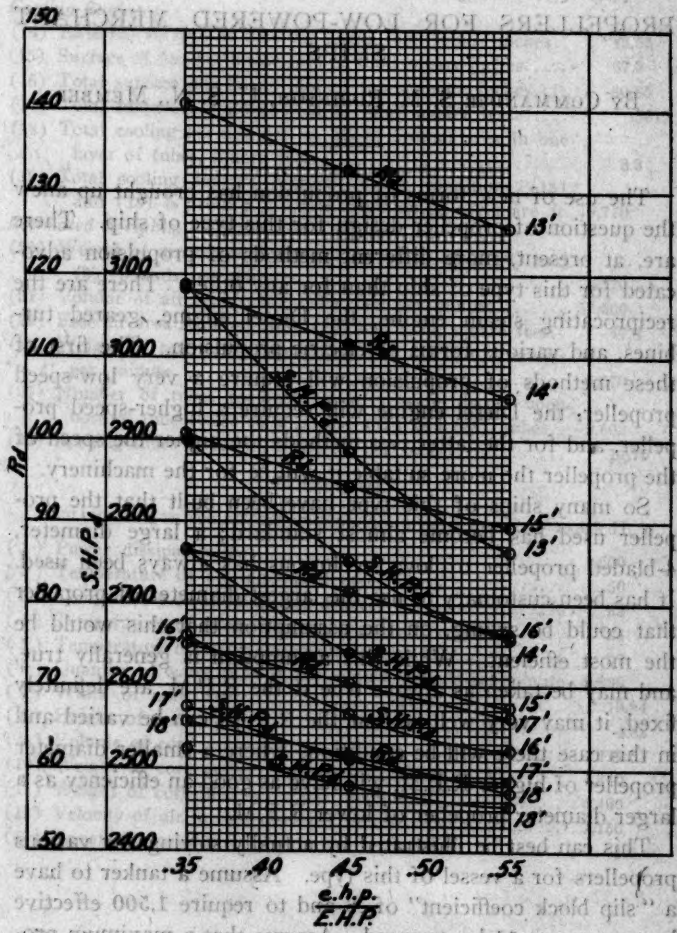

150
140
130
120
110
100
90
80
70
60
50
3100
3000
2900
2800
2700
2600
2500
2400
S.H.P.
R1
13'
14'
15'
16'
17'
18'
.35
.40
.45
.50
.55
e.h.p.
E.H.P.

FIG. 1.

D	18	18	18	17	17	17	16	16
$\frac{e.h.p.}{S.H.P.}$ (Assumed)	.35	.45	.55	.35	.45	.55	.35	.45
e.h.p.	1500	1500	1500	1500	1500	1500	1500	1500
S.H.P. = e.h.p. ÷ $\frac{e.h.p.}{S.H.P.}$	4286	3333	2730	4286	3333	2730	4286	3333
v	11	11	11	11	11	11	11	11
$\frac{V}{v}$ (Sheet No.22 for $\frac{e.h.p.}{S.H.P.}$) (Curve B)	.67	.71	.74	.67	.71	.74	.67	.71
$V = v \div \frac{V}{v}$	16.43	15.50	14.87	16.43	15.50	14.87	16.43	15.50
C	2.491	2.491	2.491	2.491	2.491	2.491	2.491	2.491
D^2	324	324	324	289	289	289	256	256
$\frac{E.A.}{D.A.} \times E. \sigma_p = \frac{S.H.P. \times C}{D^2 \times V}$	2.005	1.653	1.412	2.247	1.854	1.585	2.257	2.093
S.B.C.	.8	.8	.8	.8	.8	.8	.8	.8
$\frac{E.A.}{D.A.}$ (3 blades) (Sheet No. 24)	.246	.22	.20	.254	.225	.214	.237	.223
$(\frac{E.A.}{D.A.})_4$ (4 blades) = $\frac{4}{3} \times \frac{E.A.}{D.A.}$	.328	.2933	.2666	.338	.3125	.2853	.3163	.2973
P.C. (Sheet 20 for $(\frac{E.A.}{D.A.})_4$)	.666	.685	.697	.652	.675	.689	.651	.661
I.H.P. = $\frac{S.H.P.}{P.C.}$	6435	4870	3915	6570	4940	3965	6790	5060
S.H.P. = I.H.P. x .92	5920	4480	3600	6040	4545	3647	6250	4645
10^3 (Sheet 21 for $\frac{e.h.p.}{S.H.P.}$)	2.984	2.297	1.864	2.984	2.297	1.864	2.984	2.297
$S.H.P_p$ = S.H.P. ÷ 10^3	1985	1950	1932	2025	1977	1957	2093	2020
K (Sheet 19 on $C_2 - C_3$)	1.27	1.27	1.27	1.27	1.27	1.27	1.27	1.27
$S.H.P_d$ = $S.H.P_p$ x K	2522	2478	2454	2573	2510	2485	2660	2565
$V_1 = \frac{P.S \times (1-S)}{V \times 101.33}$	16.066	14.118	12.494	17.306	15.221	13.454	18.767	16.577
P = (D. x V_1) ÷ V_1	18.40	19.77	21.44	18.14	17.52	18.55	14.0	14.96
S (Sheet L for $\frac{E.A.}{D.A.}$ and S.B.C.)	.055	.054	.053	.056	.055	.054	.059	.058
v^V (Sheet 21)	4824	5451	5254	4824	5451	5254	4824	5451
v^V (Sheet 21)	1325	1325	1325	1325	1325	1325	1325	1325
$s = \frac{S. \times (S.H.P_d) \times V^V}{(S.H.P.) \times v^V}$	.0765	.0825	.0885	.0779	.0837	.0899	.0806	.0868
I - s	.9235	.9177	.9117	.9221	.9163	.9101	.9194	.9132
$P_d = \frac{v \times 101.33}{P (I - s)}$	65.6	61.4	57.1	78.0	70.8	66.1	86.6	81.8

TABLE I.

16	16	16	15	15	15	14	14	14	13	13	13
.35	.45	.55	.35	.45	.55	.35	.45	.55	.35	.45	.55
1500	1500	1500	1500	1500	1500	1500	1500	1500	1500	1500	1500
4005	3855	3730	4365	3855	3730	4005	3855	3730	4295	3855	3730
11	11	11	11	11	11	11	11	11	11	11	11
.67	.71	.74	.67	.71	.74	.67	.71	.74	.67	.71	.74
16.45	15.50	14.87	16.45	15.50	14.87	16.45	15.50	14.87	15.45	15.50	14.87
2.491	2.491	2.491	2.491	2.491	2.491	2.491	2.491	2.491	2.491	2.491	2.491
256	256	256	225	225	225	196	196	196	169	169	169
2.857	2.093	1.796	2.886	2.398	2.088	3.51	2.738	2.263	3.845	3.17	2.705
.0	.0	.0	.0	.0	.0	.0	.0	.0	.0	.0	.0
.287	.258	.229	.314	.274	.240	.346	.308	.271	.387	.336	.300
.3825	.3373	.3063	.4190	.3663	.3307	.4615	.4025	.3615	.516	.449	.400
.651	.661	.679	.608	.645	.668	.679	.617	.645	.544	.668	.619
6790	5050	4080	7080	5190	4105	7400	5410	4285	7080	5470	4410
6230	4645	3700	6465	4775	3775	6810	4975	3895	7245	5215	4085
2.984	2.297	1.864	2.984	2.297	1.864	2.984	2.297	1.864	2.984	2.297	1.864
3095	2380	1985	2175	2079	2025	2285	2165	2090	2428	2270	2175
1.27	1.27	1.27	1.27	1.27	1.27	1.27	1.27	1.27	1.27	1.27	1.27
2660	2665	2522	2764	2640	2573	2900	2750	2655	3085	2885	2745
.787	16.677	14.784	20.447	17.986	16.284	22.868	19.762	17.796	24.421	21.707	19.649
16.0	14.96	16.1	13.05	13.95	13.78	10.88	10.97	11.70	8.74	9.20	9.84
.088	.085	.084	.089	.086	.085	.088	.089	.086	.087	.091	.089
3984	3681	3234	4284	3681	3234	4284	3681	3234	4284	3681	3234
1325	1325	1325	1325	1325	1325	1325	1325	1325	1325	1325	1325
0908	.0838	.0899	.0821	.0884	.0915	.0875	.0884	.0882	.0880	.0931	.0945
9184	.9162	.9101	.9179	.9146	.9085	.9124	.9116	.9069	.9070	.9069	.9055
96.6	81.8	76.1	100.7	94.3	89.3	118.8	111.8	105.0	140.7	132.5	126.8

BLE I.

will lie between .35 and .55, so we will assume values of .35, .45, and .55 for this ratio and then solve for diameters of 18, 17, 16, 15, 14, and 13 feet. Table No. 1 gives the complete solution for all these screws.

The data given in Table No. 1 has been plotted in Figure 1. From this Figure, the points of equal S.H.P. (equal efficiency) for the various diameters can be found; these have been plotted in the form of curves of equal S.H.P. in Figure 2.

From these two Figures, it will be seen that the 18 feet diameter propeller is unsatisfactory, even for a reciprocating engine as the R.P.M. are too low. For this type of engine, a diameter of 17 feet, 16 feet, or 15 feet would be satisfactory and the same efficiency could be obtained with any one of the three, provided there are no limitations on the R.P.M.; for the 17 feet diameter the R.P.M. would be 75, for the 16 feet diameter the R.P.M. would be 82, and for the 15 feet diameter the R.P.M. would be 89.5 and the S.H.P. would be 2,575 in all three cases.

If a Diesel engine is to be used, higher R.P.M. would be desired and it will be necessary to make some sacrifice in efficiency, if the 2,575 S.H.P. curve is taken as the standard. However, for any given R.P.M. within the limits of Figure 2, the smaller diameters will give the higher efficiency. For example, suppose it is desired to have the Diesel engine make 120 R.P.M. If a diameter of 13.4 feet is used, it will require 2,765 S.H.P.; if a diameter of 13.78 feet is used, it will require 2,865 S.H.P. and there will be a loss in efficiency of $\frac{2,865-2,765}{2,765}$ or 3.617 per cent.

If a mechanical reduction gear or electric propulsion is used, the R.P.M. become of still greater importance as it is desirable that the propeller speed be as high as possible without too great a loss in efficiency. Suppose, for example, that in choosing the type of machinery to be used, it was decided that a loss of propeller efficiency of between 6 and 7 per cent

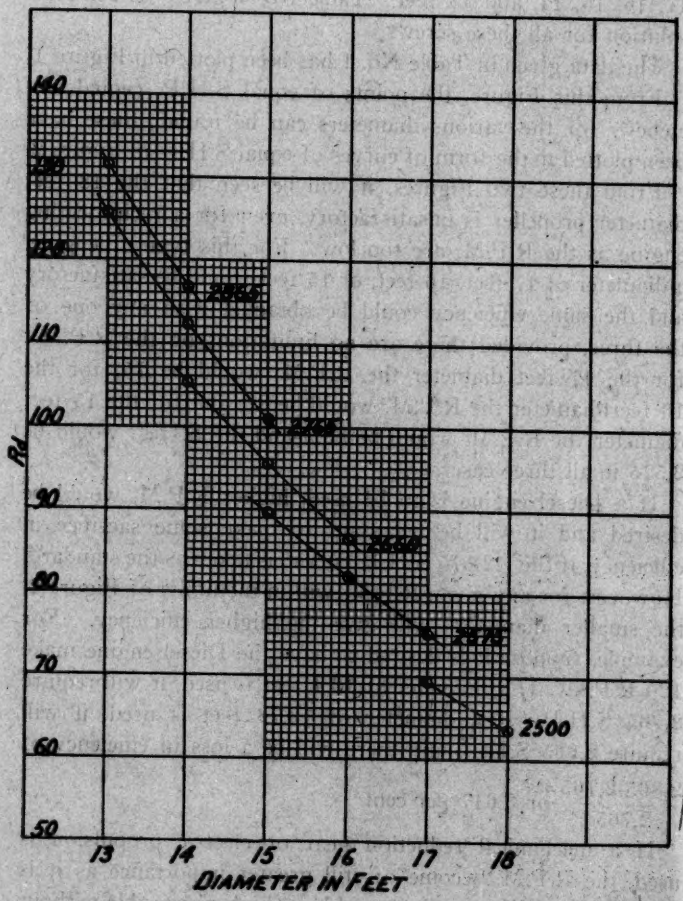

140
110
100
90
80
70
60
50
R
2500
13
14
15
16
17
18
DIAMETER IN FEET

FIG. 2.

would be the maximum allowed for this type of machinery as compared with a reciprocating engine. This would give 2,765 S.H.P. as the horsepower required for propulsion. By referring to Figure 2, it will be seen that if we use a propeller 15 feet in diameter, the R.P.M. will be 100.7 and if we use a propeller 13 feet in diameter, the R.P.M. will be 125.5 or an increase of about 25 per cent. This difference is of the greatest importance for either of these types of machinery. For example, if a geared turbine, running at 3,600 R.P.M., were used, it would give a reduction of 36 : 1 in one case and 29 : 1 in the other case; if electric propulsion were used, in the case of an induction motor running from a 50-cycle circuit, it would mean a 60-pole motor in one case and a 48-pole motor in the other case and the difference in the diameters of the two motors might be sufficient to make it impossible to install a 60-pole motor while the 48-pole motor could readily be put in; also from the standpoint of the motor itself, the 48-pole motor is much better.

It must be understood that these conclusions in regard to propellers are not general, but apply only to the specific case under discussion—that is, small power with large diameter propellers; in the case of large powers and high speeds, there are many other considerations that enter into the propeller problem and these make the above conclusions inapplicable.

ELECTRIC STEEL FURNACES AND ELECTRIC STEEL.

With Especial Reference to Steel Castings and the Greaves Etchells Type of Electric Furnace.

By Commander F. J. Cleary, U. S. N., Member.

In preparing the following article data has been obtained from various sources:

"Iron Age," "Marine Engineering," New York, "Engineering," London, etc. First hand information was gained from the installation and operation of the 6-ton Greaves Etchells furnace at the Navy Yard, Mare Island.

I am especially indebted to Mr. W. A. Lynn, Expert Electrical Aid, Navy Yard, Mare Island, for assistance rendered in preparing the description of the electrical equipment, and in collecting operating and cost data.

GENERAL.

Making steel in the electric furnace cannot be considered a new process, but the rapid expansion of this process, especially since the outbreak of hostilities in 1914, has been unprecedented.

In 1908, there was only one furnace in the United States producing electric steel—and this furnace produced only 55 tons of electric steel during that year. In 1910, there were 10 electric furnaces in operation. On January 1, 1918, the number of electric furnaces in the United States had increased to 233, with an estimated output for 1917 of not less than 500,000 tons. Reports indicate that during 1918 and the first six months of 1919 at least 100 additional electric furnaces have been erected and put in operation.

The trade journals report that in the United Kingdom, at the end of 1918, there were in operation 117 electric steel

furnaces with a nominal monthly output of 31,256 tons. The electric steel industry has consequently been firmly established on a commercial basis.

While the first electric arc furnace was constructed by Sir William Siemens in 1878, it was not until twenty years later that the first successful electric furnace, from a commercial standpoint (the Stassano furnace) was placed in operation. It is unquestionable that the development of the electric furnace and the production of electric steel were greatly retarded by the relatively high cost of electric power. The truly remarkable advances made in generating and distributing electric power in the last fifteen years, with its consequent reduction in cost, together with the superiority of the electric furnace in certain respects has resulted in very rapid development in the production of electric steel.

While electric furnaces are extensively used in producing steel castings from steel scrap, they are still more extensively used in making special alloy steel, high speed and carbon tool steel, and for melting the "ferro alloys" which are constituents of the steels having special chemical and physical properties.

Before the outbreak of the war in 1914, the greater part of the special alloy steels and the ferro alloys used in the United States was imported. The complete stopping of the supply of special steels and ferro alloys from abroad compelled the steel makers in the United States to produce these materials. While the production of the special steels was attended by many difficulties and frequent early failures, the steel makers continued their efforts until complete success was achieved. The special steels now produced in the United States have higher and more uniform physical characteristics than the steels formerly imported from Germany. At present the greater part of the ferro alloys used in the United States are of domestic manufacture. The more important of these alloys are ferro manganese, ferro silicon, ferro tungsten, ferro vanadium, ferro tinanium, ferro nickel, ferro chrome, etc.

The electric furnace is also used for producing low phosphorous pig iron, but not to any large extent.

Electric furnaces are especially adapted for the production of special alloy steels used in the construction of rifles, light and heavy artillery and other ordnance material, automobiles, airplanes, etc. The experience of the past five years has shown conclusively that the electric furnace produces steel that is superior to open hearth steel and fully equal to crucible steel in quality. On account of the greatly increased cost of crucibles, due to the war, it is claimed that the electric steel process is more economical than the crucible process.

One of the most noticeable features in the development of the electric furnace is the increase in size of the units. While a few years ago the size of electric furnaces was limited to two or three tons capacity, furnaces of five and six tons capacity are now common, and furnaces of ten, fifteen, twenty-ty-five, and even thirty, tons capacity are in use.

Electric furnaces, whether used for heat treatment or for melting and refining metals, may be divided into three general classes—induction furnaces, resistance furnaces and arc furnaces.

While electrical resistance is the cause of the heat used in each class of electric furnace, the form of resistance varies in the different classes.

In induction furnaces the heat is generated in the metallic charge by means of induced currents. The induction furnace is seldom seen in steel plants but is largely used in melting non-ferrous metals.

In resistance furnaces, the heat is generated either by the current passing through the metallic charge or through a resistance forming part of the furnace itself. The first type is known as the direct resistance type. The second type is known as the radiation type. The resistance furnace is used principally for heat treatment and annealing; it is also adapted for melting non-ferrous metals.

In arc furnaces, the heat is generated by the current passing from one electrode to another sometimes passing through the charge as well. The arc furnace is used almost exclusively in producing electric steel. The number and arrangement of electrodes differ widely.

There are two types of arc furnaces—the arc radiation type and the arc resistance, or arc conduction type. The Stassano furnace, which has been previously referred to as the first successful electric furnace from a commercial standpoint, was of the arc radiation type and melted the metal almost entirely by radiation from the electric arc. The number of furnaces of the arc resistance type far exceeds those of the arc radiation type.

The arc resistance type melts the metal by the heat generated by the electric arc and by the passage of the electric current through the charge, the current passing from one electrode to another either along the surface of the charge or through the charge. In melting steel, alternating current is used practically to the exclusion of direct current as furnaces of the arc resistance type installed for direct current have not been successful. The exact cause of this failure of the direct current type is not definitely known, but one theory advanced is that the effort of the carbon is to flow from one electrode (the cathode), the pure iron remaining around the other electrode, and that this action is strong enough to throw the carbon out of solution in the bath. The current may be either one, two or three phase; three phase is generally used for the reason that a balance is more easily obtained. Inasmuch as the arc is a form of electric resistance, the voltage between electrodes depends upon the length of the arc; the usual difference of potential between electrodes is about 100, but arrangements are provided to vary the voltage between small limits to provide for the changing conditions while melting. The voltage required for melting in the furnace is stepped down by transformers from the transmission lines.

METHODS OF OPERATING.

Electric steel is made by one of two methods—either by charging the cold metal in the electric furnace where it is both melted and refined, or by melting the cold metal by the open hearth or the Bessemer process and transferring the molten charge to the electric furnace where it is refined only. Sometimes both methods are combined.

The method of refining from a molten charge is especially adapted for furnaces of ten tons capacity or greater—so far the cold melt method has not been successful in large units.

The cold melt method is especially adapted for furnaces of less than ten-ton capacity largely due to the much simpler equipment required and the much better control of the chemical analysis.

For steel furnaces intended to produce castings it is especially important that the furnace be so installed and equipped that it may be operated with the utmost simplicity and convenience, so that the time lost between heats for patching the lining and the pouring spout, recharging the furnace, adjusting or replacing electrodes, etc., may be reduced to a minimum.

Electric furnaces may be either acid or basic, but in the United States the basic lined electric furnaces outnumber the acid lined. While it is claimed that the acid furnace is simpler, cheaper and faster to operate, the reason for the preference for the basic furnace is apparent enough; with the acid bottom, little or no sulphur or phosphorous can be removed from the charge, and if a steel low in sulphur and phosphorous is desired, the charge must be carefully selected with a consequent high cost of raw material—with the basic bottom the sulphur and phosphorous can be eliminated down to negligible quantities (not to exceed .015 per cent), by the use of a basic slag, permitting the use of cheap raw material; in addition, the percentage of carbon can be readily increased or decreased as desired and the alloys can be added to the bath with very little loss.

ADVANTAGES.

While electric steel is perhaps chemically purer than steel produced by any other process, it does not necessarily follow that electric steel has superior physical characteristics. A poor grade of steel can be, and has been, produced in the electric furnace. It has been proved, however, that the electric furnace when properly designed and handled is capable of producing steel that cannot be excelled either in chemical or physical characteristics, and experience has shown that when made to the same chemical analysis, electric steel averages about 15 per cent greater either in tensile strength or ductility (depending upon its heat treatment). Comparative tests of open hearth and electric steels show the following physical characteristics:

	C.	*Tensile Strength.* *Pounds.*	*Elongation in 2 in.* *Per cent.*
Open hearth	.12	56,500	29.82
Electric	.12	65,200	26.05
Open hearth	.20	58,320	28.35
Electric	.20	73,150	22.75

An interesting series of comparative tests was made of steel rails made from open hearth and from electric steel. Test pieces were taken from open hearth and electric furnace steels that had practically the same physical characteristics at normal temperatures. These test pieces were subjected to tests at much lower temperatures. At 40 degrees F. the electric furnace steels showed decidedly better tests.

One great advantage of the electric furnace is that the composition of the steel can be much more exactly controlled than in any other process of making steel and this exact control is of vital importance in the production of special alloy steels. This exact control also insures more certain results in heat treatment and the steel may be subjected to a higher degree

of heat without injury. Electric steel is also more resistant to shock. These excellent characteristics are due to the fact that the steel is made by the clean heat of the electric arc in the reducing (deoxidizing) atmosphere of a closed furnace, and is thus protected from the contamination of fuel or gases of combustion, resulting in the steel being more solid and more free from gases. When properly made, electric steel lies dead in the ladle and in the molds, and being lower in sulphur and more fluid it is much less liable to show shrinkage cracks, pipes and blow holes. Another advantage is that the charge in the electric furnace requires less excess alloy to insure the proper amount in the finished steel.

In open hearth furnaces, losses in manganese and chrome may run as high as 40 per cent of the contained elements as charged in the furnace.

The claim has been made by some makers of electric steel that it is necessary to use only one-half as much ferro-manganese as would be required by the best open hearth practice. Another advantage claimed is that the alloy additions may be made in the furnace instead of in the ladle, thereby insuring thorough incorporation and a more homogeneous steel, although it must be pointed out that the adding of the alloys in the furnace has the disadvantage of requiring more of the alloy than if they are added in the ladle. Another advantage of the electric furnace is that it may be used in recovering alloy scrap (where such recovery is attempted in the open hearth furnace a large part of the alloy scrap is lost in the slag).

Another advantage of the electric furnace is its flexibility—one heat a day can be melted and the furnace allowed to cool with very little damage to the lining. If necessary five, six, or even seven, heats per day can be melted.

The temperature in the electric furnace depends upon the electric energy put into it. This is easily regulated on the automatic regulation by the setting of the rheostats provided

with the furnaces, the amount of electric energy being accurately registered by meter readings.

As electric steel can be made very hot there is very little loss from sculls in the ladles. Whether the metal is hot enough to pour is determined by the very simple test of taking a spoonful of metal from the furnace and pouring it out of the spoon.

A spherical spoon about five inches in diameter is used, the spoon being carefully slagged up over the bath. The spoonful of steel is taken from the center of the bath and carefully poured over the lip of the spoon at a slow even rate. The temperature of the steel is shown by its fluidity and by the amount of scull left in the spoon. If the entire spoonful pours freely the bath is at the proper temperature to insure perfect castings of small section and practical absence of scull in the ladle.

For the purpose of comparison with the method of handling and the results obtained in the Greaves Etchells furnace at Mare Island, the following description of handling a 6-ton Heroult electric furnace, copied from one of the trade journals, will be of interest.

"The process may be divided into two periods:

(1) Melting and oxidizing.

(2) Recarburizing and reducing (deoxidizing).

It appears paradoxical to remove carbon during one period and to add it at another, but this is necessary in order to oxidize the phosphorous. Furnaces for reducing phosphorous are lined with basic material, such as magnesite or dolomite. Limestone is charged into the furnace with the scrap. The rust on the scrap furnishes the oxygen and sometimes ore or roll scale is charged to augment this supply. The slag formed oxidizes the phosphorous which then combines with the lime, forming calcium phosphate. Carbon, silicon, manganese and sulphur are also oxidized to a greater or less extent by this slag.

11:50 A. M. Material charged:

Boiler plate	5,980 pounds.
Stampings	5,991 pounds.
Limestone	700 pounds.

12:20 P. M. Completed charging (current switched on).

3:20 P. M. Charge melted down.

Preliminary analysis under black slag:

Carbon	.06
Silicon	.014
Sulphur	.032
Phos.	.009
Mang.	.08

3:40 P. M. The oxidizing (black) slag is now poured and skimmed off as clean as possible to prevent re-phosphorizing and to permit adding carburizing materials. For this purpose carbon is added in the form of powdered coke, ground electrodes or other forms of pure carbon.

The deoxidizing slag is now formed by additions of lime, coke and fluor spar (and for some analyses, ferro-silicon). The slag changes from black to white as the metallic oxides are reduced by these deoxidizing additions and the reduced metals return to the bath. A good finishing slag is creamy white, porous and viscous. After the slag becomes white, some time is necessary for the absorption of the sulphur in the bath by the slag to take place.

The action of desulphurization in the electric furnace is as follows: After the bath has been oxidized as previously described, the slag acts on the metallic sulphides, eliminating them by forming calcium sulphide, which is taken up by the slag. The white slag disintegrates to a powder when exposed to the atmosphere and has a pronounced odor of acetylene when wet. Further additions of recarburizing material are

added as needed to meet the analysis. The further reactions are shown by the following:

3:40 P. M. Recarburizing material added.

Ground electrodes	130 pounds.
Ferro manganese	25 pounds.

Analysis:

Carbon	.75
Silicon	.011
Sulphur	.03
Phos.	.008
Mang.	.26

To form white slag there was added:

Lime	225 pounds.
Powdered coke	75 pounds.
Fluor spar	55 pounds.

4:50 P. M.

Analysis:

Carbon	.75
Silicon	.014
Sulphur	.012
Phosphorus	.008
Manganese	.28

(Note the reduction of the sulphur content.)

During the white slag period the following alloying additions were made:

Pig iron	500 pounds.
Ferro silicon	80 pounds.
Ferro manganese	9 pounds.
6 per cent carbon ferro chrome	146 pounds.

The furnace is now rotated forward to an inclined position and the charge poured into the ladle, from which, in turn it is poured into molds.

5:40 P. M. Heat poured.

Analysis:

Carbon	.97
Silicon	.25
Sulphur	.014
Phos.	.013
Mang.	.33
Chromium	.70
Ingot weight poured	94.0 per cent.
Scull	2.7 per cent.
Loss	3.3 per cent.

Total current consumption for the heat, 4,700 kilowatt-hours or 710 kilowatt-hours per ton."

ELECTRODES.

One of the most important items of cost of operation of the electric furnace is the cost of electrodes. Not only in the amount consumed per ton of steel in the ladle, but also in the cost of wastage incurred in renewing them. The amount consumed per ton of steel produced cannot be very greatly reduced. The cost of wastage can be very greatly cut down by careful handling. The electrodes used in electric furnaces are either amorphous carbon or graphite. While the electric conductivity per unit of cross section of graphite electrodes is three times that of amorphous carbon and their use would permit of smaller holes in the roof, amorphous carbon is generally preferred for two reasons.

First, the relative cost—the present cost of graphite electrodes is 29.6 cents per pound, of amorphous carbon, 8 cents per pound.

Second, the thermal conductivity per unit of cross section of graphite is nearly six times that of amorphous carbon, so that for the same current-carrying capacity the drainage of

heat from the furnace by graphite electrodes is double that by amorphous carbon electrodes.

To obtain the required total electric conductivity the diameter of amorphous carbon electrodes must consequently be greater than if graphite electrodes are used. This, although adding to the weight, is of value because of the umbrella effect of the greater diameter protecting a much greater area of the roof from the intense heat radiated directly from the arc. The number of electrodes required depends upon the number of phases used and whether or not there are electrical connections through the bottom of the furnace. With bottom electrical connections, a single phase requires one electrode; two-phase, two electrodes, and three-phase, two or four electrodes (except where bottom connection is to a middle point). If no bottom connections are used and the current passes from one electrode to another, a single phase requires two electrodes; two-phase, four electrodes, and three-phase, three electrodes.

ELECTRODE CONSUMPTION.

The following electrode consumption has been claimed by various authorities for types of electric furnaces with gallows arm type of electrode control:

Snyder (single phase), 7 pounds per ton of steel in ladle.

Heroult, Greaves-Etchells, Electro Metals (English), 13 pounds per ton of steel in ladle.

Furnaces having no gallows arm:

Stobie (English), 6.5 pounds per ton of steel in ladle.

(The above figures have not been verified, but they appear to be very low.)

ELECTRIC FURNACE LININGS.

The chief materials used for electric furnace linings are magnesite, dolomite and silica.

In an acid furnace, silica is used for lining hearth, walls and roof.

In a basic furnace, the lining of the hearth up to a point a little above the slag line is usually magnesite, although dolomite may be used. A dolomite lining is less affected by extreme temperature conditions than a magnesite lining. The walls above the slag line may be lined with magnesia, silica or chrome brick depending upon working conditions, although silica brick is preferred, as it is cheaper and stands up better under the intense heat. The magnesite brick being basic and the silica brick being acid, the usual practice is to put one or two layers of chrome (neutral) brick between. The roof of a basic furnace is lined with silica brick. As silica bricks have a large coefficient of expansion and crack and chip easily, that part of the lining made of silicious material should be handled with especial care. Usually the pouring spout is lined with firebrick, as this material is not so susceptible to changes of temperature as either magnesite or silica brick.

Actual practice has demonstrated that a lining about 9 inches thick is the most economical. An 18-inch lining was found to last only 20 per cent longer than a 9-inch lining.

A roof 9 inches thick should last 70 heats or more, its life being largely dependent upon the skill and care of the operators. A spare roof should always be on hand, so that a damaged roof may be replaced immediately.

MAKING BOTTOM.

The bottom may be made by any one of several different methods.

In one method a paste is made of magnesite with heated tar and rammed into the bottom of the furnace with hot tampers. When starting the first heat with such a bottom, the bottom must be covered with heavy pieces of scrap. The first heat will take up carbon from the tar in the bottom to such an extent that it is advisable to cast it into pigs and use it up as scrap.

A second method is to ram in several layers of a mixture of dolomite and slag, burning in each layer with the arcs from the electrodes. This is accomplished by laying pieces of electrodes on the bottom to provide a path for the current from one electrode to another. This method makes an excellent bottom, but the heat that burns in the bottom is hard on the roof which softens and sometimes spalls, largely dependent upon the skill of the operator. Another method is to line the bottom with three layers of magnesite brick, filling in the crevices and covering the bricks with a half-inch layer of powdered magnesite. When this bottom is burned in, a very even surface is obtained.

As the Greaves-Etchells furnace is of a special design, it was necessary to make several departures from standard practice in lining the roof, walls and bottom. A description of the method of lining this furnace follows:

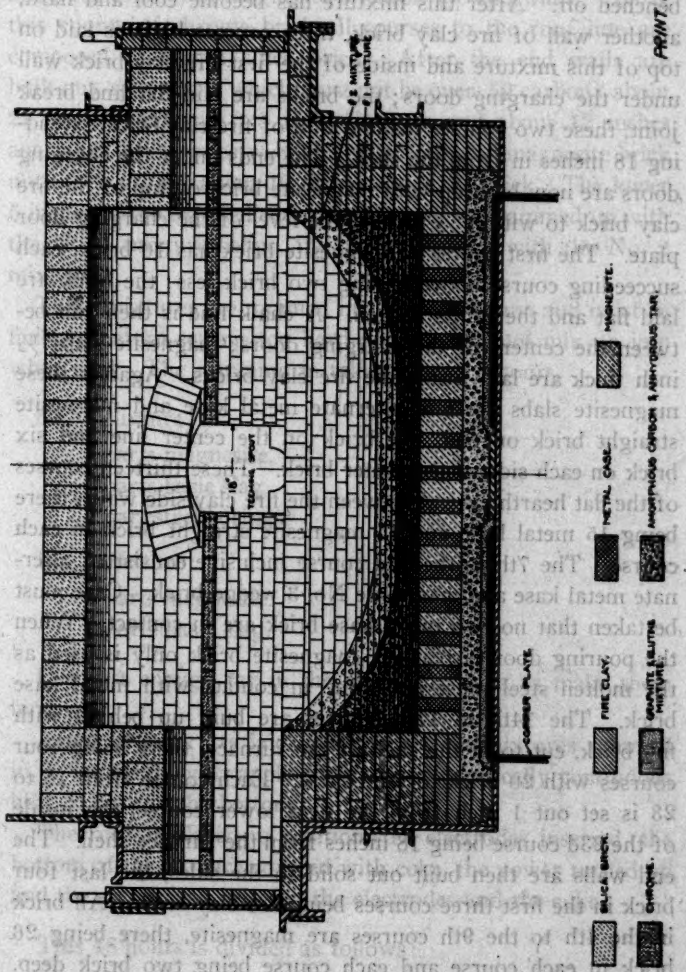
LINING THE ROOF.

A form was made of boiler plate curved to the contour of the roof on which to build up the roof. This form is mounted on wheels so that it may be pulled out under the crane when a new roof is to be put on the furnace. The spare roof is kept on this form (Photo 1). The roof frame is placed on the form and the roof started. Building one-half of the roof up to the top of the arch only will be described, the other half being a duplicate. Two 6-inch × 9-inch silica brick are cut to fit the corners; the first course consisting of 28-9-inch straight silica brick set on end is then laid. As the first course is laid on the side flange of the roof frame, ¼ inch of silica sand is spread over the form to raise the other courses of brick up to the level of the first course. Courses two to ten consist of 9-inch straight silica brick, 32 brick to the course, all joints being broken, the end bricks being cut to suit. Course eleven consists of No. 1 end wedge brick. Course twelve consists of 9-inch straight brick. Course thirteen con-

sists of 27-9-inch straight brick, this being the first course abutting the bricks encircling the electrode ports, which are outlined by a sheet metal form. The electrode ports are encircled with 33 Nos. 2 and 3 silica arch brick, the outside diameter of these brick circles is 24 inches. Course fourteen consists of 25-9-inch straight brick. Courses fifteen to eighteen each consists of 20-9-inch straight brick. Course nineteen consists of 27 No. 1 wedge brick. Courses twenty to twenty-three each consist of 32-9-inch straight brick. Course twenty-four consists of 27 No. 1 wedge brick. Course twenty-five consists of 27 No. 1 wedge brick. This course is the key course and is not laid until both sides of the roof are laid. All brick must be properly lined up, using a straight edge as necessary. After all the bricks are laid a thin coating of silica cement is swept over the top of the roof. The metal forms outlining the electrode ports are now removed. The water-jacketed electrode cooling collars are placed over the electrode ports, the circle of brick around the ports being built up as necessary to provide a flat surface for the cooling collars.

LINING THE BOTTOM, SIDE WALLS AND END WALLS. (Print 1).

Fire clay brick are cut to fit the bottom of the furnace at the ends directly under the two charging doors. The first brick is laid flat in the exact center and ten bricks are laid flat on each side of the center brick. Three more courses are laid flat on top of this first course, the joints of each course being broken. Seven hundred and sixty pounds of graphite and 50 gallons of glutrin are then thoroughly mixed and warmed on a hot plate and when hot and rather stiff, rammed in on top of the copper plates to a depth of two inches and extending to 2 inches beyond the copper plates on each side. Then amorphous carbon (broken to walnut size) and anhydrous tar are mixed to a stiff consistency and rammed in on top of the graphite and glutrin mixture to a depth of 6 inches for a distance of 15 inches on each side of the center line and


PRINT 1.
2 MIXTURE
1 MIXTURE
MAGNESITE.
METAL CASE.
AMORPHOUS CARBON & ANHYDROUS TAR.
TAPPING SPOUT
FIRE CLAY.
GRAPHITE & SILICA MIXTURE.
BASIC BRICK.
CHROME.

from there this mixture tapers to about one inch in thickness at the edges of the graphite and glutrin mixture and is there benched off. After this mixture has become cool and hard, another wall of fire clay brick, three courses high, is laid on top of this mixture and inside of the first fire clay brick wall under the charging doors; the bricks are laid flat and break joint, these two courses make a wall of fire clay brick extending 18 inches in from the shell. The ends under the charging doors are now built up with magnesite brick on top of the fire clay brick to within ½ inch of the level of the charging door plate. The first course of magnesite brick has 16 brick, each succeeding course above having two brick less; the brick are laid flat and the joints broken. A chalk line is then run between the centers of the charging doors; magnesite slabs ½ inch thick are laid inside the fire clay bricks. Against these magnesite slabs are laid alternate metal kase and magnesite straight brick on end, one brick on the center line and six brick on each side of the center brick. These thirteen courses of the flat hearth extend between the fire clay side walls, there being 15 metal kase and 15 magnesite straight brick in each course. The 7th to the 23d course inclusive consist of alternate metal kase and magnesite No. 3 wedge brick. Care must be taken that no two metal kase brick are in contact. When the pouring door is reached, magnesite brick only is used as the molten steel must not come in contact with metal kase brick. The 14th to 17th courses are built up behind with fire brick, cut to fit the shell of the furnace, there being four courses with 20 brick to the course. Each course from 18 to 23 is set out 1 inch from the next lower course, the inside of the 23d course being 18 inches from the furnace shell. The end walls are then built out solid to the sides, the last four brick in the first three courses being fire clay brick. All brick in the 4th to the 9th courses are magnesite, there being 26 brick in each course and each course being two brick deep. The level of the charging door has been reached on the 9th course.

The 24th course of the side walls is a neutral line of chrome brick and extends all around the furnace, tying into the 10th and 11th courses of magnesite brick on the end walls. Above this course of chrome brick all courses to the roof line are composed of 9-inch silica brick. After the end walls are built out, steel rods (which must not be over .50 carbon) about 22 inches long and 3/8 inch diameter and about 12 inches apart, are driven in between the metal kase and magnesite brick until they project about 10 inches above the brick. The lower 5 inches of the exposed ends of these rods are rammed up with the No. 1 mixture below, the upper 5 inches with the No. 2 mixture.

These mixtures are rammed in 1 inch at a time and must be finished before setting too hard as they will not mix or join when cool. The ramming in takes about 15 hours.

No. 1 mixture:

5 parts magnesite.
2 parts basic slag.
1 part steel borings.

No. 2 mixture:

5 parts magnesite.
2 parts basic slag.

Use just enough tar in the above mixtures to make them bind well.

After all brick are laid, and the bottom mixtures rammed in, it is necessary to apply 48 hours of practically continuous heat to burn in the bottom.

The roof is placed in position, the electrodes inserted, the bottom of the furnace covered with coke, the power turned on and the arcs struck between the electrodes and the coke.

This 48 hours is divided as follows:

8 hours heat.
2 hours shut down.

5 hours heat.
1 hour shut down.
7 hours heat.
1 hour shut down.

The cycle is then repeated.

The shut down periods prevent overheating and permit the heat developed to soak in, thereby preventing spalling, which is very apt to occur with such relatively poor heat conductors as furnace refractories.

In applying this heat the side walls and especially the silica roof must be watched to see that they do not overheat and come down. If any sign of overheating develops, the power must be shut off and not applied again until the overheated roof has had a chance to cool sufficiently.

The first steel heat produced will absorb so much carbon from the bottom that it is advisable to cast it into pigs and recharge it into the furnace in small quantities divided over several heats.

The refractory bottom material originally purchased with this furnace was known under the trade-name "Magnite." It was received at the Navy Yard, Mare Island, several months before it could be used, and, due to exposure to the air, it slacked and became worthless. This material was replaced by dead burnt magnesite, which is much superior to "Magnite" for the reason that it is not susceptible to air slacking. This material was purchased from the White Rock Mines in Napa County, California, and is much superior to the Austrian spater magnesite previously used in large quantities in the United States. It was purchased under the following specifications:

Silica (soluble), not over 6 per cent.
Lime, not over 1 per cent.
Iron and aluminum, 4 to 6 per cent.
Ignition loss, not over 1 per cent.
Magnesia, not less than 87 per cent.

The material purchased complied with the following chemical analysis:

Silica (soluble), 5.4 per cent.
Silica (insoluble), 3.5 per cent.
Iron and aluminum oxide, 4.8 per cent.
Ignition loss, trace.
Magnesium oxide, 86.3 per cent.

This material cost $35.00 per ton.

THE GREAVES-ETCHELLS FURNACE.

The Electric Steel Furnace in operation at the Navy Yard, Mare Island, is of the Greaves-Etchells type of six long tons rated capacity, although eight tons can be melted and poured by building up the pouring door.

It was purchased under Washington N. P. O. Contract 206, Series 1918, under the following specifications:

"One 6-ton electric furnace.

The furnace to have four vertical electrodes, a conducting hearth and inside dimensions—9½ × 7½ feet. The equipment to include complete switch gear with main line breaker, step-down transformers for a three-phase line, automatic regulators, flexible leads and bus bars from furnace to transformers. Lay out to be in accordance with contractors' blue prints. Tilting mechanism and motor to tilt furnace. Four electrode operating motors (two on rear and two on front, all conveniently located).

The Government is to furnish all foundations and platforms for apparatus and also transformer room. The contractor is to furnish the services of an expert engineer to superintend erection of the furnace complete and to instruct the Government operators in the operation of the equipment for a period of thirty (30) calendar days.

The Government is to furnish 2,300 volts 3-phase, 60 cycles, on tap at the transformer room and also 230 volts direct current for branch circuits on tap at transformer room.

The Government is also to furnish sufficient water at furnace for electrode water coolers. All equipment to be designed to the above power characteristics.

The Government will provide suitable siding facilities for the unloading and delivering of the furnace to the building in which it will be installed and also crane facilities and operators for getting the furnace into position."

The above specifications included the following equipment and supplies not listed in detail therein: Ammeters, wattmeters, switchboards, cables and copper necessary for the distance of 15 feet from the transformers, electric regulators, motors and switch gear. Also necessary material for the first lining and sufficient brick for backing up the lining and original roof.

The total contract cost of the electric furnace was:

Contract price	$47,500.00
Allowed contractors for installing	5,960.00

The weight of this furnace is about 60 tons.

In addition, sufficient material for two complete linings plus 5 per cent as per the following list was purchased under Bureau Contract No. 39,391, with the Harbison-Walker Refractories Co., at a total cost of $10,354.00.

	Quantity.	*Material.*		*Unit Price.*	
(*a*)	2250	W. Star Silica	9″ Straights	$208.20	per M
(*b*)	600	W. Star Silica	9″ No. 1 Wedge	208.20	per M
(*c*)	90	W. Star Silica	Large 9″ Straights	.317	each
(*d*)	22	W. Star Silica	Large 9″ Wedge	.296	each
(*e*)	210	W. Star Silica	9″ No. 2 Arch	208.20	per M
(*f*)	74	W. Star Silica	9″ No. 3 Arch	208.20	per M
(*g*)	2470	W. Star Silica	9″ Straights	208.20	per M
(*h*)	84	W. Star Silica	9″ No. 2 Wedge	208.20	per M
(*i*)	380	W. Star Silica	9″ No. 1 Wedge	208.20	per M
(*j*)	442	Chrome	9″ Straights	901.00	per M
(*k*)	4200	Magnesia	9″ Straights	801.00	per M
(*l*)	1120	Magnesia	9″ No. 3 Wedge	801.00	per M
(*m*)	284	Magnesia Metalkase	13½″ Straights	1.00	each

(*n*)	852	Metalkase	9" Straights	.751 each
(*o*)	1180	Metalkase	9" No. 1 Key	.751 each
(*p*)	380	Metalkase	9" Jambs	.851 each
(*q*)	380	Metalkase	13½" Jambs	1.10 each
(*r*)	3424	Franklin Crown	9" Straights	369.35 per M
(*s*)	4480	lbs. D. B. Magnite in sacks, at .0625 per lb.		140.00 per long ton
(*t*)	4480	lbs. Furnace Magnite in sacks, at .067 per lb.		150.00 per long ton

In installing the furnace the following features were provided. The ladle pit was waterproofed so as to keep it free from water or excessive dampness, and was provided with a non-slip surface cover which could be readily and quickly removed. The motor pit was provided with a removable cover. The ground around the furnace was laid with a brick floor 4½ inches thick.

POWER HOUSE STANDPOINT.

The available electric power and the price at which it can be obtained are considerations of great importance in contemplating the installation and the size of an electric furnace.

The capacity of the transformers required per ton of steel melted decreases as the furnace capacity increases, due to the fact that the radiation losses are greater in proportion in the smaller furnace. The K.W.H. per ton also decreases as the furnace capacity increases.

In considering the relation of the electric furnace to the power house and to the cost of power, there are several important features to consider.

First—As all power is now sold on a load factor* basis the more continuously the furnace can be operated, the smaller will be the proportion of the readiness to serve charge to the energy charge, which means that the average cost per K.W.H. will be less. Consequently the size of the furnace should bear such a relation to the tonnage output desired as to insure

* The load factor is the percentage of average load to peak load.

practically continuous operation of the furnace. The load factor on a single furnace operating practically continuously is about 35 per cent—when operating two furnaces under similar conditions the load factor rises to 50 per cent, which compares favorably with other loads carried by the power plant. A furnace in continuous operation is kept hot and no electric energy is expended in heating a cold furnace—consequently the furnace efficiency will be higher, which makes the saving per K.W.H. all the more important.

Second—The furnace should be operated with the least possible loss of time between heats. During the time between heats the user loses a part of his demand time and increases the proportion of the readiness to serve charge. The furnace is also cooling very rapidly, and the electrodes are oxidizing more rapidly than when in active use.

Third—The furnace should operate at the highest practicable power factor* which can be reached without undue disturbance of the power plant load. Operating at a high power factor reduces the electrode, transformer, line and generator losses. The power factor of well designed furnaces ranges from 85 per cent to 90 per cent, but it is necessary to have a large reactance of 10 per cent or 12 per cent in the system to choke the unavoidable fluctuations of current in the arc. Melting cold scrap makes a very severe demand on the power equipment as it is subjected to sudden momentary overloads of from 50 per cent to 75 per cent and the transformers must be designed for such an overload.

ELECTRIC CURRENT CONSUMPTION.

The electric furnace cannot be regarded as a cheap melter, its value lying more in its capacity for refining.

The current consumed per ton of steel produced by the electric furnace depends upon operating conditions such as kind of steel produced, degree of refining, capacity of furnace,

* The power factor is the ratio of the actual watts to the apparent watts.

whether melting and refining from a cold charge or refining only from a molten charge and upon the number of heats produced per day. The current consumption per ton is greater for small furnaces than it is for large furnaces. If a grade of scrap is used relatively high in sulphur and phosphorous, more slag, made principally of lime, must be melted, increasing the current consumption. In a furnace melting and refining certain grades of plain carbon steel and using one slag, the current consumption will vary from 500 to 650 K.W.H. and may be as little as 450 K.W.H. per net ton of steel produced. In a furnace producing tool or alloy steels from a cold charge and using two slags the current consumption may vary from 700 to 800 K.W.H. per net ton of steel produced. When refining from a molten charge, the current consumption is relatively low; in furnaces operating on what is known as the triplex system, the current consumption is as low as 110 K.W.H. per ton. The power and time required to refine a molten charge is about one-third that required to melt and refine a cold charge. Consequently, the electric furnace operating on molten charges has three times the capacity of the same furnace operating on cold charges. The thermal efficiency of the modern electric furnace is about 65 per cent, so that a large decrease in the K.W.H. per ton of steel produced cannot be looked for. Consequently any considerable improvement in electric furnace efficiency will result from more reliable mechanical features and more durable refractories especially those used in lining the roof.

TRANSFORMERS.

These are a most important part of the furnace installation and should receive the best possible care. All transformers for electric furnaces are built to order for the particular furnace and its electrical equipment. Consequently, the breakdown of a transformer, unless a spare is on hand, would shut down the furnace for several weeks if not months.

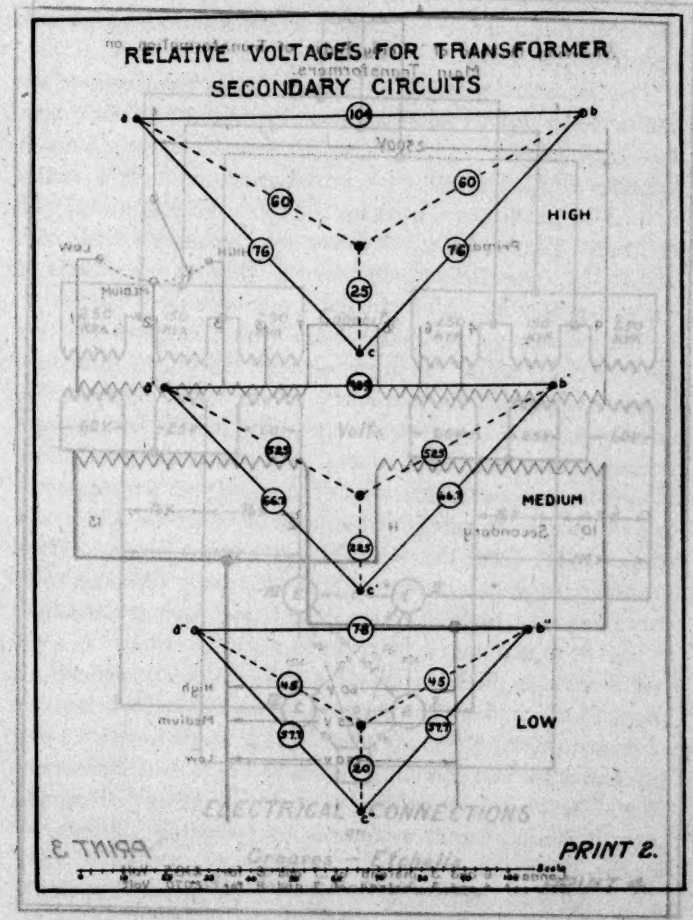
THE ELECTRICAL EQUIPMENT OF THE GREAVES-ETCHELLS SIX-TON ELECTRIC FURNACE.

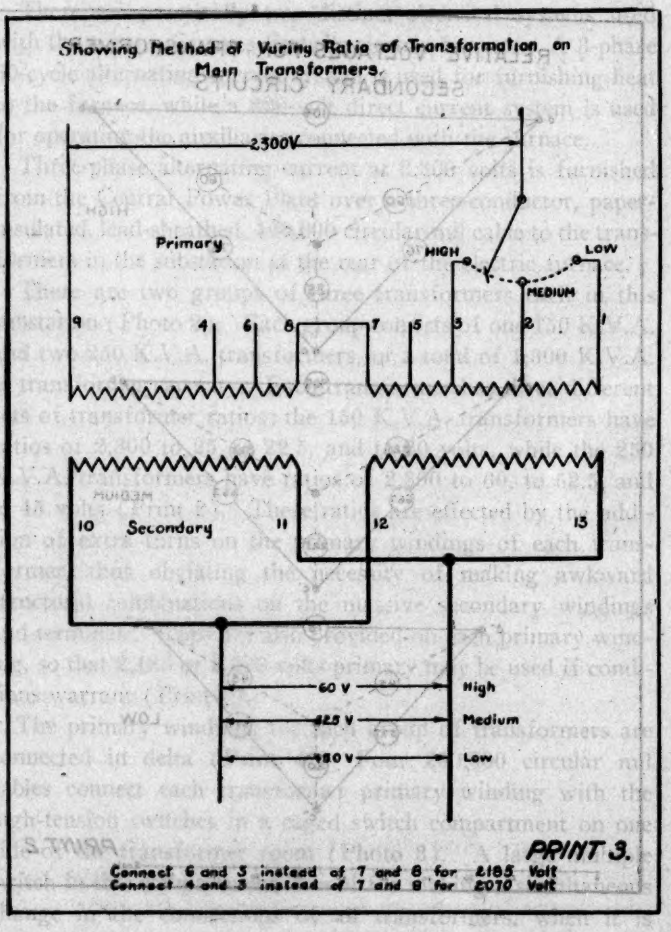
There are practically two distinct electrical systems used with the six-ton Greaves-Etchells electric furnace. A 3-phase 60-cycle alternating current system is used for furnishing heat to the furnace, while a 220-volt direct current system is used for operating the auxiliaries connected with the furnace.

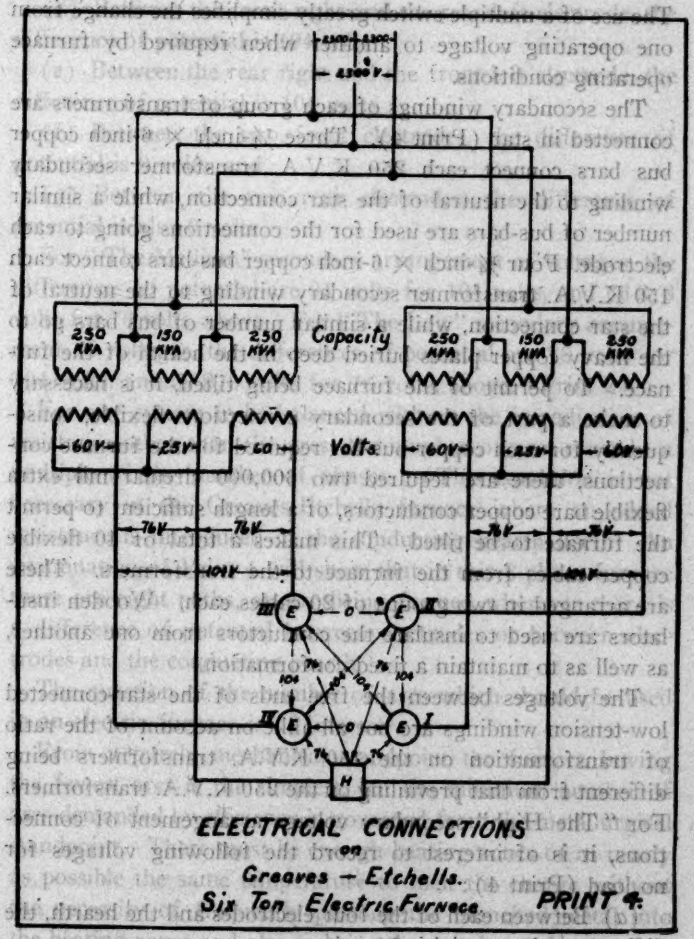
Three-phase alternating current at 2,300 volts is furnished from the Central Power Plant over a three-conductor, paper-insulated, lead-sheathed, 400,000 circular-mil cable to the transformers in the substation at the rear of the electric furnace.

There are two groups of three transformers each in this substation (Photo 2). Each group consists of one 150 K.V.A. and two 250 K.V.A. transformers, or a total of 1,300 K.V.A. in transformer capacity. Each transformer has three different sets of transformer ratios; the 150 K.V.A. transformers have ratios of 2,300 to 25, to 22.5, and to 20 volts, while the 250 K.V.A. transformers have ratios of 2,300 to 60, to 52.5, and to 45 volts (Print 2). These ratios are effected by the addition of extra turns on the primary windings of each transformer, thus obviating the necessity of making awkward structural combinations on the massive secondary windings and terminals. Taps are also provided on each primary winding, so that 2,185 or 2,070 volts primary may be used if conditions warrant (Print 3).

The primary windings for each group of transformers are connected in delta (Print 4). Four 250,000 circular mil cables connect each transformer primary winding with the high-tension switches in a caged switch compartment on one side of the transformer room (Photo 3). A large multiple switch in this compartment is used for making a simultaneous change in the connections of all transformers, when it is desired to change from "The High" to "The Medium," or to "The Low" voltage on the secondary terminals. The handle of this multiple switch extends through the brick wall


RELATIVE VOLTAGES FOR TRANSFORMER SECONDARY CIRCUITS
104
60
60
HIGH
76
76
25
c
MEDIUM
225
225
225
c'
78
45
45
LOW
c''
PRINT 2.


Showing Method of Varing Ratio of Transformation on Main Transformers.
2300V
Primary
HIGH
LOW
MEDIUM
9
4
6
8
7
5
3
2
1
10
Secondary
11
12
13
60 V
High
425 V
Medium
450 V
Low
PRINT 3.
Connect 6 and 3 instead of 7 and 8 for 2185 Volt
Connect 4 and 3 instead of 7 and 8 for 2070 Volt



ELECTRICAL CONNECTIONS
ON
Greaves — Etchells.
Six Ton Electric Furnace. PRINT 4.

separating the transformer room from the furnace room, where it is conveniently located within reach of the operators. The use of a multiple switch greatly simplifies the change from one operating voltage to another when required by furnace operating conditions.

The secondary windings of each group of transformers are connected in star (Print 4). Three ¼-inch × 6-inch copper bus bars connect each 250 K.V.A. transformer secondary winding to the neutral of the star connection, while a similar number of bus-bars are used for the connections going to each electrode. Four ¼-inch × 6-inch copper bus-bars connect each 150 K.V.A. transformer secondary winding to the neutral of the star connection, while a similar number of bus bars go to the heavy copper plates buried deep in the hearth of the furnace. To permit of the furnace being tilted, it is necessary to make a part of the secondary connections flexible, consequently for each copper bus-bar required for the furnace connections, there are required two 600,000 circular mil extra flexible bare copper conductors, of a length sufficient to permit the furnace to be tilted. This makes a total of 40 flexible copper cables from the furnace to the transformers. These are arranged in two groups of 20 cables each. Wooden insulators are used to insulate the conductors from one another, as well as to maintain a fixed conformation.

The voltages between the free ends of the star-connected low-tension windings are not all alike on account of the ratio of transformation on the 150 K.V.A. transformers being different from that prevailing on the 250 K.V.A. transformers. For "The High" secondary voltage arrangement of connections, it is of interest to record the following voltages for no load (Print 4):

(*a*) Between each of the four electrodes and the hearth, the difference of potential is 76 volts.

(*b*) Between each of the two electrodes on the right side, the difference of potential is 104 volts.

(*c*) Between each of the two electrodes on the left side, the difference of potential is 104 volts.

(*d*) Between the front right and the rear left electrode, the difference of potential is 104 volts.

(*e*) Between the rear right and the front left electrode, the difference of potential is 104 volts.

(*f*) Between the two front electrodes, the difference of potential is 0 volts, and

(*g*) Between the two rear electrodes, the difference of potential is also 0 volts.

For "The Medium" secondary arrangement of voltages, the differences of potential are 91 volts for 104 volts, and 66 2/3 volts for 76 volts, above. For "The Low" secondary arrangement of voltages, the differences of potential are 78 volts for 104 volts, and 57 2/3 volts for 76 volts, above, (Print 2).

For any system to be balanced without the introduction of resistance other than the arcs, the number of arcs must be a multiple of the number of phases. (This condition is not necessary in the Greaves-Etchells furnace where an added resistance is introduced in the conducting hearth itself. An examination of Print 4 will show that in this 3-phase furnace there are eight paths, an arc being formed wherever there is a difference of potential between electrodes or between electrodes and the conducting hearth.)

The question of the number of arcs which should be used in an electric furnace is of interest.

From a purely mechanical standpoint the furnace having the fewest arcs is the simplest, but the minimum number of arcs demanded by efficiency is governed from the metallurgical standpoint. There must be enough heating zones of as nearly as possible the same temperature to melt the charge without the necessity of frequently pushing the unmelted metal into the heating zones and also to bring the metal and the slag to a high and uniform temperature. In the basic furnace the slag is largely lime and the greater the proportion of lime the greater the melting point of the slag—it is only possible to

have a fluid high basic slag with a high furnace temperature; consequently the number of arcs must be sufficient to ensure a high uniform temperature throughout the furnace. In the Greaves-Etchells furnace the eight major paths for the flow of current between the various electrodes and the hearth, shown on Print 4 and tabulated on page 270, show the complete manner in which this particular arrangement of electric circuits is employed for melting the metal. As the ohmic resistance of a hotter metal is always greater than a colder metal, the preponderance of current in any one of the major circuits will tend to flow through the colder regions, other conditions remaining the same. This physical phenomenon has the tendency of equalizing the temperature throughout the various sections of the bath. This uniformity of temperature also insures a uniformity of the chemical and metallurgical processes taking place and hence a uniformity of the physical properties of the final product.

Before describing the other electrical devices used with this installation, the difficulties usually encountered in the operation of an electric furnace, used for melting steel scrap and similar material, will be referred to. At the beginning of a heat, the charge is of a very heterogeneous character and the flow of current through any one electrode of a very decidedly erratic nature, so much so, that the other electrodes are also affected. As an electrode burns its way through the scrap metal, many short-circuits of varying degrees of severity occur, the most severe occurring when a large piece of metal becomes dislodged from the top of the pile and rolls down and jambs in between the electrode and the walls of the hole formed by it in the charge. As soon as the pools of molten metal are formed at the tips of the electrodes, the most violent of the surges become less frequent and soon cease altogether.

It will be observed that the transformer groups are independent of one another, except that the primary systems are connected to a common supply circuit and that the secondary systems are connected to the furnace and its electrodes. Any

abnormal condition which may arise in the secondary system will have an effect upon the primary system. For instance, if the furnace is cold, the hearth is non-conducting, and while there will be a difference of potential between the hearth and any one electrode, there will be no current flowing until the hearth attains a certain temperature. The result will be that two of the transformers will carry all the load, and the current on the supply lines will become unbalanced. Inasmuch as there are two groups of transformers, the unbalancing will not be as excessive as it would be in case only one group was used. The use of a delta system of primary connections also has the effect of distributing the unbalance upon all three incoming supply lines. In case of a short at any electrode when the hearth is cold and non-conducting, the surge of current is at its maximum. In many systems the results are so disastrous that it becomes necessary to install choke coils in the primary circuits. This brings the power-factor very low with all its consequent disadvantages. To meet this type of surge, the inventors of this system adopted star-connected secondaries. It will be noted from Print 4 that the current has to traverse at least two sets of secondary windings of two different phases whenever a surge occurs at any particular electrode, thus reducing the strain upon a single transformer and tending to distribute the shock upon all three incoming feeders. The reactance of the transformers has been so selected that the maximum current is about twice the full-load current of one of the 250 K.V.A. transformers.

The hearth of this furnace is formed of basic material which is a conductor of the second class, the resistance decreasing as a function of the rise in temperature. When the furnace is operating, the surface of the hearth in contact with the bath is about 1800 degrees C. At this temperature the resistance of the hearth at the surface is very small, but increases rapidly toward the bottom of the lining. This condition has been taken advantage of to provide for a flow of current to and through the hearth. During ordinary operation the resist-

ance of the hearth has a decided effect upon limiting the magnitude of the current during an incipient surge at any electrode. In this case, the effect of a surge does not affect the operation of the other electrodes to the same extent as in the case of a non-conductive hearth, principally because the surplus current passes through the hearth and not through any of the other arcs and electrodes.

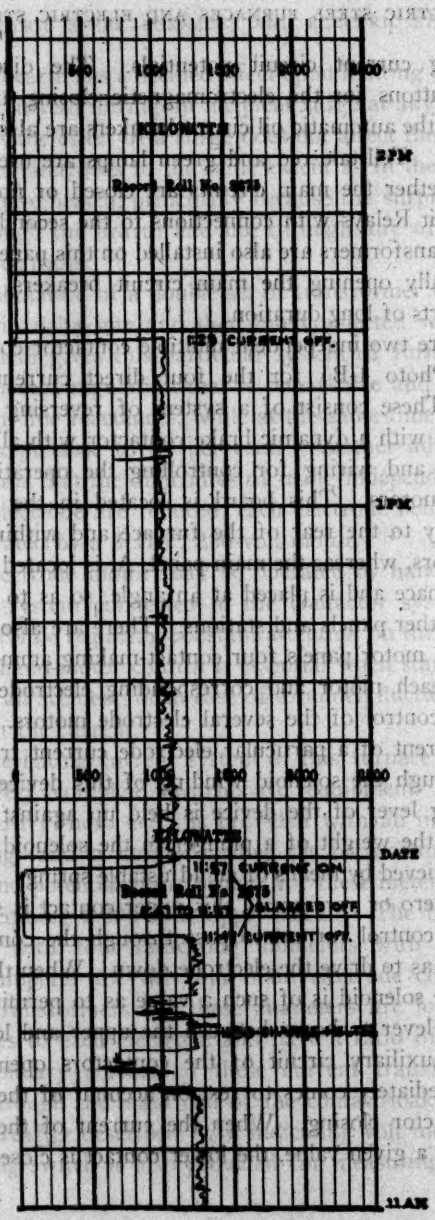
The combination of a double set of transformer banks each connected in delta-star and then interconnected with a conductive hearth gives this particular system of furnace operation an inherent regulation that is remarkable and avoids the use of main-line reactances, while at the same time the power factor is kept close to 90 per cent. Another advantage of this system is that the difficulties of using independent motors for each electrode are reduced, each electrode being satisfactorily operated by a 3 H.P. 230-volt motor.

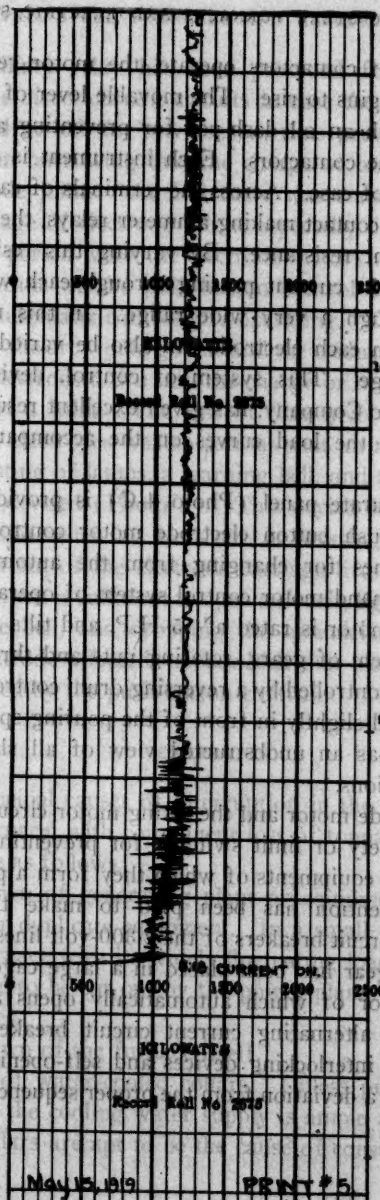
Each electrode motor may be operated by hand, or by an automatic system connected to the contactor system of the motors. While requiring constant attention on the part of the operator, hand-motor control is essential when starting up, or in case of an accident, or in case of a sudden fluctuation of the load, or of the current.

The alternating current system of this furnace is provided with a very complete switchboard equipment. The main switchboard (Photo 4-A) is provided with an indicating, an integrating, and a curve-drawing set of wattmeters. The current and potential transformers for these meters are placed in the incoming 2,300-volt supply circuit. The board is also provided with four indicating ammeters with a range of 0-8,000 ampères for the four main electrode circuits. The current transformers for these ammeters are located in the main lines to each electrode and have a ratio of 1,600 to 1. The secondary current of these current transformers also passes through the main coil of the contact-making ammeter relay, hereafter described. An indicating volt meter with the usual plug switch is also provided for measuring the various

alternating current circuit potentials. The direct current control buttons for the electromagnetic closing and opening system of the automatic oil circuit-breakers are also located on this panel. Tell-tale red and green lamps are used for indicating whether the main circuits are closed or not. Inverse Time Limit Relays with connections to the secondary system of main transformers are also installed on this panel board for automatically opening the main circuit breakers in case of severe shorts of long duration.

There are two independent multiple contactor control panel boards (Photo 4-B) for the four direct current electrode motors. These consist of a system of reversing contactors interlocked with a dynamic brake contactor with all necessary resistances and wiring for controlling the operation of the electrode motors. This board is located in the brick wall immediately to the rear of the furnace and within reach of the operators, whereas the main panel, A, is located to the left of the furnace and is placed at an angle, so as to be viewed from the other panels and stations. There are also placed on these D. C. motor panels four contact-making ammeter relays (one for each motor and corresponding electrode) for the automatic control of the several electrode motors. The secondary current of a particular electrode current transformer passes through the solenoid winding of this device. A contact-making lever of the device is held up against an upper contact by the weight of a plunger in the solenoid, although partially relieved by means of an adjustable spring. When the current is zero or very small, this upper contact is such as to permit the control current to pass through the contactors in such a way as to drive the electrode down. When the current through the solenoid is of such a value as to permit the contact of the lever to hover between the upper and lower contacts, the auxiliary circuit of the contactors open and the motor immediately comes to rest on account of the dynamic brake contactor closing. When the current of the solenoid coil exceeds a given value, the lower contact is closed and the


KILOWATTS
RPM
Record Roll No. 2618
FIELD CURRENT OFF.
KILOWATTS
DATE
11:57 CURRENT ON
Record Roll No. 2618
11:57 CURRENT OFF.
TIME


KILOWATTS
Record Roll No. 2575
9 AM
KILOWATTS
Record Roll No. 2575
May 15, 1919
PRINT # 5

reversing set of contactors operate, the motor reverses and the electrode begins to rise. The movable lever of this device is provided with an oil dash-pot for preventing a too rapid operation of the contactors. Each instrument is enclosed in a glass dustproof case. Across the terminals of each solenoid winding of the contact making ammeter relays, there is placed a variable shunt resistance. By varying this resistance, the magnitude of the current passing through each winding can be varied through a very wide range. In this manner the current through each electrode can also be varied through a very wide range. This system of control, devised by the General Electric Company, has given excellent results, as may be noted from the load curves on the accompanying sheet (Print 5).

A small separate panel (Photo 4-C) is provided for the four sets of push button electrode motor controls, and the multiple switches for changing from the automatic motor control to the hand motor control system of operation.

The tilting motor is rated at 15 H.P. and tilts the furnace through a system of gears, rotating nuts and threaded rods. This motor is controlled by a reversing drum controller located to the right and slightly in front of the pouring spout, so that the operator has an unobstructed view of all slagging and pouring operations.

Each electrode motor and the tilting motor circuits are provided with safety or limit switches for preventing injury to the mechanical equipments of which they form a part.

Especial attention has been paid to make the multiple switches and circuit breakers of the 2,300-volt lines safe. The entire switch gear has been placed in a large cage, the opening of the door of which automatically opens all switches therein. The alternating current circuit breakers are also provided with interlocking devices and self-opening switches in case there is a deviation from the proper sequence of switching operations.

An auxiliary relay has been provided for automatically opening the direct current circuits in case the alternating current should go off for any reason, thus preventing the electrodes from burrowing into the charge or hearth and breaking electrode joints. At first there was considerable trouble with breakage of electrode joints, especially during the initial heating-up period when the electrode motors were being operated by means of the push-button control system, but also when the automatic control was operating. There were occasions when the arc would go out and the electrode motor would force the electrode down into the charge or hearth causing the breakage of a joint. The loss of time involved in replacing the broken electrodes was so serious that an annunciator system, consisting of lamps, a warning bell, and a 4-drop annunciator was devised (Photo 5). This device depends upon the principle that when an electrode is pressing heavily on its tip, the bottom of the electrode carriage lifts from its track. This movement is sufficient to open a circuit, lighting the lamps, ringing the bell, and dropping the annunciator of the particular electrode affected, upon which that electrode is immediately lifted by the push-button control.

OPERATION.

The principal causes of interruption in the operation of an electric furnace, with consequent loss of tonnage and increase of cost, are as follows:

Breakdown of generating plant or equipment and lack of stand by or duplicate equipment. In a small power plant the electric furnace with its widely fluctuating loads is usually the first service to be cut out by the power plant when any emergency arises.

Breakdown of the furnace itself.

Failure of water-cooled collars. Unless properly designed and unless the cooling water supply is ample and reliable, the cooling collars are apt to be the cause of constant trouble and

delay. After considerable trouble at Mare Island, due to this cause, the best installation was found to consist of steel collars with generous cooling spaces, a separate pipe and hose connections for each collar, a visible indication that the water is being supplied and a duplicate source of water supply, so that in case one source of water supply fails, the other source is immediately available.

Electrodes slipping through holders—this is a question of proper design of electrode clamps and proper operation in setting them up correctly.

Electrodes breaking—this is a question of proper design of automatic electrode control and of safeguards and alertness on the part of the operating personnel.

The best charge for the electric furnace consists of heavy scrap, averaging about 30 pounds, mixed with about 20 per cent of short heavy steel turnings and punchings. The hearth should be covered with a layer of turnings or punchings about 3 inches thick, then about 12 inches of the heavy scrap and that covered with 2 or 3 inches of turnings and punchings. The scrap should not be heaped up in the middle of the furnace. As this is a basic furnace all foundry scrap should be tumbled or otherwise cleaned of the silica (acid) sand before charging in the furnace, otherwise the acid sand will attack the basic lining. Part of the lime (slag), about 75 pounds, should be charged with the metal. When the furnace is charged to a level about 6 inches above the level of the charging door sills, lower the electrodes and turn on the current, using the intermediate voltage.

Proper handling of the electrodes is one of the most important features of electric furnace operation. When first starting the furnace, if the automatic electrode motor control is used, the electrodes are apt to be forced down on the cold charge, canting them and breaking the electrode through the threaded connectors. For this reason the hand-motor control is used when starting up the furnace. The hand control is used until the ammeters steady down to about 4,000 ampères,

which will take about 30 minutes. Then switch in the high voltage. (In usual practice the low voltage is not used at any time unless it is desired to hold back the heat). After a few minutes' operation under the high voltage, if the ammeters show fairly steady readings and everything is satisfactory, shift over to the automatic control. The electrodes should be watched carefully at all times. The switchboard must be watched to see that both direct current and alternating current circuits are operating properly. If the direct current goes off, the electrode motors stop and the electrodes will remain stationary, causing injury both to the furnace and to the metal. In this case the clutches should be operated, throwing the electrode motors out and the hand wheels in and the electrodes operated by the hand wheels. If the alternating current goes off, the operator should shift from the automatic to the hand motor control and raise the electrodes so that a sudden surge of current will not result from the alternating current coming on again without warning. When the alternating current returns, shift to the automatic control and watch the electrodes until the furnace is again running smoothly. When working up to the first slag, carry about 4,500 ampères on each electrode. Owing to the fact that the heat of the furnace expands the chain electrode clamps and does not expand the electrodes proportionately, particular attention must be paid to properly setting up the electrode clamp screws. If this is not done, the electrodes are apt to slip through the clamp. If this takes place, shift at once from automatic to hand motor control and switch off the power, otherwise the electrode will burn a hole in the bottom of the furnace. The electrode clamp should then be tightened up and the electrode raised free of the bath before turning the power on, otherwise the sudden surge of current may throw out the circuit breakers in the power house, causing further delay. If the end of an electrode breaks off in the furnace, every effort must be made to remove it at once. Not only will it delay the heat,

but it will run up the carbon content of the bath, resulting in great trouble and delay in working it down again.

When the electrical load has become fairly steady and the automatic electrode control is operating, the maximum possible electrical energy, within the capacity of the transformers, should be put into the furnace. The steady current per electrode should never exceed 5,500 ampères and should be maintained as close as possible to 4,500 ampères. As the metal melts it collects in the middle of the hearth and forms a small bath. In about thirty minutes from the time of turning on the current, the electrodes will have burned through the mass of scrap so that a steady arc will be maintained between the electrode and the liquid metal.

The first slag consists mainly of lime. In a six-ton furnace about 75 pounds of pure burnt limestone is used for the first slag; 20 pounds of fluor spar are added as the first slag forms to thin it. Care should be taken that just sufficient iron oxide is present either in the form of rust on the scrap or by adding iron ore or mill scale with the first slag to effect the complete combustion of the carbon as carbon monoxide gas. If the scrap is clean and free from rust, it may be taken as a general rule that for every 0.1 per cent of carbon in the scrap, 58 pounds of iron ore, or mill scale, will be necessary in the first slag. (At Mare Island the steel scrap averaged 0.34 per cent in carbon, and, when the scrap was very clean, 200 pounds of hammer scale was added to the charge. Usually the scrap was well rusted and no scale was needed.) The amount of iron ore or hammer scale to be used with clean scrap is determined as follows:

At the rate of 0.1 per cent carbon, six tons of scrap would contain twelve pounds of carbon, and in order to remove the carbon as CO, sixteen pounds of oxygen would be required.

$$C + O = CO$$
$$12 + 16 = 28$$

The usual form in which oxygen is added is magnetite (Fe_3O_4).

$$Fe_3 \quad O_4 = Fe_3O_4$$
$$56 \times 3 + 16 \times 4 = 232$$

Hence, fifty-eight pounds of magnetite will furnish the required sixteen pounds of oxygen.

Mill scale can usually be considered as equal to magnetite. Hammer scale is practically the same, except that it may not be as pure as mill scale. In a recent chemical analysis of the hammer scale used in the electric furnace, it was found to contain eighty-seven hundredths of one per cent of carbon in addition to an excess of iron over what would be found in magnetite.

An analysis of a sample of representative steel plate, such as is used in the construction of vessels at Mare Island, showed the carbon content to be 0.3 per cent.

The average carbon content of the charge is estimated as follows:

	Per cent of Total Charge.	*Per cent of Carbon (Average).*	*Extensions. Per cent.*
Foundry Scrap	25	0.28	0.07
Turnings	15	0.60	0.09
Plate Scrap and Punchings	60	0.30	0.18
Average carbon content of one charge			0.34

On a basis of fifty-eight pounds of scale for each 0.1 per cent of carbon in the charge, the total amount of scale required would be approximately two hundred pounds.

In operating the furnace the carbon is reduced to between 0.04 per cent and 0.06 per cent at the end of the decarbonizing period and this is the usual range of values obtained on the first preliminary test.

The melting should be continued as fast as possible. When the charge is completely melted, the heating should be con-

tinued at the maximum voltage for about twenty minutes to insure that the metal will be hot enough not to freeze up while slagging off. The melter must use his own judgment at this stage as to what additions he makes to the slag, because in order to get the best results it will be necessary to have the slag so that it is fluid enough to run off the steel by itself when the furnace is tilted, but the slag must not be too fluid as it would cut a deep groove all around the lining of the furnace. The bath should now be searched with a rabble to be certain that there are no unmelted lumps of steel on the bottom.

A carbon analysis of the bath is now obtained by pouring some of the molten metal into an open steel mold about 2 inches × 4 inches × 1½ inches, resting on a chill (Photo 6). This test piece is broken across and the approximate carbon judged from the grain of the fracture. If the carbon is under .25 per cent, the content can be judged within one or two points by the fracture and no more exact analysis is necessary for steel castings. If above .25 per cent, as in producing high carbon or alloy steel, there is greater chance of error and a check analysis should be obtained by sending drillings from the sample to the laboratory, where the carbon and manganese determinations can be obtained within 20 minutes from the pouring of the sample. After receiving this verification of the analysis, the melter is in a position to make the additions to obtain the desired carbon and manganese and other contents.

After the sample is taken from the bath and while waiting for the analysis from the laboratory, the furnace is shut down, tilted forward and the first slag, which contains part of the sulphur and a large percentage of the phosphorous of the original charge, run off into the pit, hauling the slag through the pouring door by rabbles, if necessary, care being used not to draw off any of the steel with the slag. The first slag should be removed as quickly and as completely as possible. If any of the first slag is left on the steel, the sulphur and

phosphorous that it has taken up will re-enter the bath and appear in the castings. It is not difficult to remove the first slag when high carbon steel is being made, but the low carbon steels set at such relatively high temperatures that unless the first slag is removed quickly and the second slag formed at once, the metal freezes and considerable time will be required to bring it back to the correct temperature. When all the slag has been removed and the metal is clean, the furnace is rolled back to the level position. If the carbon content is very low, carbon in the form of ground charcoal, coke, anthracite coal, or electrodes or pig iron is thrown into the bath and a part of the ferro-manganese added, causing the metal to boil, increasing the carbon content by taking up the carbon material. While the steel is bare is the time to add the carbon if the finished steel is to be relatively high in carbon. If, however, steel is desired of only ten to twenty points, it is better to carbonize at a later stage while working up the second slag. In this case, pig iron should be used, as it is almost impossible to get accurate carbonization with powdered fuel due to the difficulty of working it through the second slag.

It should be the aim of the operator to have the carbon content right when the charge is melted down, but it is preferable to have the carbon too low rather than too high, as it is much more difficult to reduce the carbon than it is to increase it. The carbon is reduced by the addition of hematite ore or preferably mill or hammer scale which oxidizes the carbon, causing the metal to boil. This is a long and tedious process which can be somewhat expedited by thrusting a water-soaked wooden pole or a green sapling into the bath to increase the boil.

The furnace is now operated on the intermediate voltage with about 4,000 amperes on each electrode. The second slag is then introduced, consisting of about 225 pounds of lime (CaO) and 20 pounds of fluor spar. (Special care should be used in purchasing a high grade of lime for this purpose. It should be well burned, the CaO should be as high as possible

and as free as possible from MgO. The analysis of the lime used at Mare Island is CaO 99.88, MgO .12.) The lime is added two or three shovelsful at a time while working up to the second slag; the special additions are added to the bath after the second slag has been completely formed and only a few minutes before the metal is ready to pour.

The second slag takes up more phosphorous and a very large percentage of the sulphur.

Finely powdered anthracite or ground electrodes should be thrown on the surface of the slag, a shovelful at a time. A reaction occurs between this carbon and the iron oxide in the slag by which carbon monoxide is set free and the iron is precipitated from the slag into the bath. When all the oxides have been thrown out of the slag the excess carbon will tend to create with the lime, under the influence of the arcs, a certain proportion of CaC which has a great affinity for any oxides, dissolved oxygen or other occluded gases in the steel.

The carbon monoxide escapes at the doorways of the furnace and around the electrodes and burns with a smoky yellow flame. The slag should be kept at the proper consistency. If the slag is too fluid, lime should be added, if too stiff a little sand should be added.

During the first stage, *i.e.*, while the metal is melting and under the influence of the first slag, the operation is de-phosphorizing and usually decarbonizing. The carbon will oxidize away first, then the manganese and the phosphorous. The phosphorous will not be eliminated from the metal until a part of the carbon has been burnt out.

During the second stage, while the second slag is being worked up, the operation is carbonizing and desulphurizing and also reducing (deoxidizing). During the first part of this stage, while being deoxidized, the metal becomes pure iron containing only very small percentages of other elements. The sulphur is only removed when the conditions are strongly reducing. Into this pure deoxidized and desulphurized iron it is possible to make additions with mathematical accuracy

so as to obtain steel with any percentage of special elements desired.

The furnace is worked at the intermediate voltage and at about 4,000 ampères on each electrode until the slag has become calcium carbide. At this stage, if all necessary additions have been made, part of the resistance is cut out on the electrode circuits until the ammeters show 5,000 ampères. This amperage is held until the bath reaches the pouring temperature.

The pouring temperature is determined by taking a small spoonful of the metal from the bath from time to time and slowly pouring it out. When the entire spoonful pours freely, the metal is hot enough.

About five minutes before the metal is ready to pour, 30 pounds (one-half) of the ferro-silicon are added to the bath—the other 30 pounds are placed in the ladle just before tapping. The silicon is used to kill the metal so that it will lie quiet in the ladle and in the molds.

The ladle is placed in position, the power turned off, electrodes raised, the pouring door opened, the furnace tilted and the metal run into the ladle followed by the slag which acts as a blanket keeping the metal hot and preventing it from sculling over.

Three pounds of aluminum (½ pound to each ton of metal) are added to the ladle when about 6 inches of metal has been poured. The aluminum is used as a deoxidizer. It takes up any oxides that may be formed in the molds and passes off as aluminum oxide, thereby reducing gas holes in the castings.

The sample of steel for final analysis is taken from the ladle a short time before the slag comes out.

The furnace is now run back, the ladle removed from the pit, the furnace again run forward and all slag remaining in the furnace pushed out through the pouring door into the pit. If necessary, a few shovelsful of dead burned magnesite

are first thrown into the furnace to take up the slag before it is pushed out of the furnace.

The charging doors are opened and the electrodes run well up. The electrodes, bottom, walls, and roof are examined. If there are any depressions in the bottom they are filled with a shovelful or two of a mixture of 80 per cent dead burned magnesite and 20 per cent of fine steel turnings and borings. The furnace is recharged, the electrodes run down to within one-half inch of the charge, and the doors closed and sealed. If the new charge is to be melted at once, the electrodes are changed if necessary. The weight of each new electrode put in use is entered on the record; the parts of the old electrode removed are carefully weighed so that an accurate record of electrode consumption may be kept. If the new charge is not to be melted at once, the water for the cooling collars is adjusted and the valve locked (to insure that the cooling water is not accidentally shut off), the switch thrown out in the transformer room and the transformer room locked so that no one may tamper with the electric installation.

As soon as the steel is in the ladle, the power house should be notified immediately so that the alternating current may be cut out.

The melter then completes the daily report giving in detail the weight and material of the charge, the weight of the additions, electrodes replaced, time of different steps, and everything that occurs during the operation of the furnace.

A copy of this report is shown on the folder.

Notes:

75 pounds of lime and 30 pounds of ground electrodes are charged with the scrap.

20 pounds of fluor spar are added as first slag begins to form to thin the slag.

Analysis under first slag—Carbon .06.

After first slag is removed, added 300 pounds pig iron, 75 pounds ferro-manganese and 225 pounds of lime (2 or 3 shovelsful at a time while working up the second slag).

Grade C. S.	No.
Metallic Charge	Wt.
Pdy. Scrap	404
Plate Scrap	401
Turnings	157
Punchings	238
Pig Iron	300
Fe. Si.	60
Fe. Mn.	75
Fe. Cr.	
Nickel	
Al	3
Total	1237

Time	Operation	Hrs
2:55	Prev. Heat Tapped	
3:30	Began Charging	
4:30	Finished Charging	
9:00	Current On	
11:20	Charge Melted	
12:25	Slagged Off	
1:39	Current Off	
1:40	First Tap	4
	Second Tap	

983 KWH per ton of

Electric Furnace Melters Report.

Mare Island Navy Yard.

Date, August 6, 1919.

8. No. of Heats On Roof 2 On Lining 78 On Bottom 78 Heat No. 78

rge	Wt.	Time	Nom. Met. Char.	Wt.
	4040			
	4015		Lime	300
	1520		Fluor Spar	40
	2580		Coke	
	300	1:10	Ground Elect.	30
	60	1:35		
	75	1:20		
	3			
	12595		Total	370

peration	Hrs.	Min.	Meter Reading	KWH	
r. Heat			524300		
ped					
an rging					
ished rging					
rent					
rge lted					
ped					
rent					
t	4	40	530200	5900	
ead					

er ton of

Chemical Analysis

Element	C.	Mn.	Si.	P.	S.	Ni.
Required	25 35	40 70	20 30	Max .04	Max .04	
Obtained						
Check						

Preliminary Test

No.	Time Sent	Time Rec.	C.	Mn.	Si.	P.	S.
1	1:00	1:20	.06	.08			
2							
3							
4							
5							

Electrode No. 1. O. K.

Electrode No. 2. O. K.

Electrode No. 3. O. K.

Electrode No. 4. O. K.

Conditions of Furnace and Remarks.

Bottom O. K.

Roof O. K.

Lining O. K.

200 lbs. of magnesite

PHYSICAL TEST		Check
Ten-Str.		
Elas. Lim.		
Elong. per cent.		
Red. per cent.		
Bend		

Ledie Conditions and Remarks

A. J. Lucas, Melter

20 pounds of fluor spar added while working up the second slag.

30 pounds of ferro-silicon added in furnace about 5 minutes before tapping.

30 pounds of ferro-silicon in ladle before tapping.

3 pounds of aluminum into ladle after about 6 inches of metal has been poured.

There is one difference between the open hearth furnace and the electric steel furnace that is frequently lost sight of. The open hearth furnace has its tap hole below the slag line so that when tapped clean metal only runs into the ladle until the end of the tap, when the slag follows, acting as a blanket to the steel underneath, keeping it hot and preventing sculls. Electric furnaces are provided with a tapping spout over which the metal and the slag pour together unless prevented by some modification of the furnace practice. One method recommended is to brick up the pouring door, leaving a sleeve brick for the metal to flow through. This sleeve brick is filled with dolomite or similar material. When the furnace is tilted, the pressure of the metal dislodges the dolomite stopper and clean metal only is tapped into the ladle. Operation at Mare Island shows that if the slag is of the proper consistency and not too viscous the slag will rise to the top of the ladle without difficulty, leaving only the clean metal underneath.

Another recommendation conducive to clean steel is to line the ladle with clean brick, rather than with ganister or brick with a fire clay wash. In Mare Island practice very satisfactory results have been obtained by lining the ladle with fire brick and then coating it with a stiff mortar made of ground fire brick and silica sand. This coating shows no disposition to flake off and go into the molds with the steel, and is very effective in protecting the fire brick lining. Holding the steel in the ladle to allow the gas and other impurities to rise to the surface is also of value, although care must be

taken that the steel is not held in the ladle long enough to become too cold to pour easily.

All precautions of this nature only assist and are useless unless the steel has been deoxidized in the furnace so that it is not wild when poured into the ladle.

Both lip pour ladles and bottom pour ladles are used. The bottom pour ladle is preferable for two reasons: one, to prevent slag from entering the molds; the other, to prevent the rapid cooling of the metal by pouring over the lip of the ladle. For these reasons the bottom pour ladle is generally used unless pouring very light castings.

When casting ingots, electric steel, because of its density should be cast in molds with refractory hot tops to prevent any possibility of pipes in the body of the ingot (Photo 7).

CARE OF FURNACE LININGS.

The furnace should be so operated that the lining may be kept at as uniform a temperature as possible, care being taken to avoid extreme cooling and extreme heating. As the lining materials have a comparatively large coefficient of expansion and contraction, if the furnace has been cold for several days it should be heated up slowly until the temperature is somewhere near the operating range.

Care should be taken to keep acid material such as sand and ganister from coming in contact with the basic lining. After a heat has been poured, care should be taken to run off all remaining slag. If any considerable quantity of slag is left in the furnace, it will coat over the hearth, then when the new charge becomes melted the heat of the molten metal will cause the old slag to attack the lining and if permitted to continue for several heats it will eat a deep hole in the lining.

In a basic furnace any slag left in the furnace should be sponged up by throwing in a few shovels of granular burnt dolomite or dead burnt magnesite as soon as the furnace is empty and rabbling it in with the soft slag or liquid steel. The

slag, steel, and magnesite will coagulate and can be pushed out of the pouring door.

The basic part of the lining should be repaired with the original material (dolomite or magnesite). Any holes should be filled up to the normal curvature of the furnace hearth. Any holes in the lining in such a position that the granular material would roll out should be filled with a mixture of magnesite and tar, put on with a long-handled shovel or a special pasting tool. The silica brick walls above the slag line and the silica roof should be repaired with a stiff mortar of two-thirds ground silica brick and one-third fire clay. Any steel that has burrowed into the lining must be dug out; this is especially the case at such points as the charging and pouring doors where steel is apt to lodge.

The average life of a roof is 100 heats, melting and refining, or about 400 working hours. The first roof usually lasts a less number of heats because it is subjected to the heat of burning in the bottom and wear and tear of changing electrodes, etc., due to inexperienced personnel. At Mare Island, the life of the first roof was 76 heats.

The life of the roof can be considerably prolonged by preventing fine dusty lime from coming in contact with it and also by keeping the top free from dust. The dust should be blown off the roof daily. When changing electrodes or doing any work over the roof, always put wooden planks on the roof and stand on them. In making electrode joints, use plenty of joint compound on the threaded connectors and screw them up tightly.

All mechanical parts of the furnace should be gone over at least once each week, tightening up bolts, bus bar clamps, electrode holders, etc. All oil cups should be kept filled and in working order.

All electrical parts of the furnace installation should be kept clean, especially from metallic pieces or metallic dust, so as to prevent short circuits. All motors should be kept in excellent operating condition. If any of the electrode motors

become inoperative, use the mechanical method of operating that electrode. Fully competent electricians only should make any necessary adjustment to the automatic regulators, relays, etc.

Do not operate the furnace above its rated electrical capacity. The K.V.A. capacity is 1,300. The current taken by each electrode should be about 4,500 amperes and should not exceed 5,500 amperes for any length of time.

The disconnecting switches located at the top and rear of the switch gear should never be opened while the current is on the furnace. They are installed only to give a positive indication that the line is clear if it should become necessary to do any work without interference with the main feeder.

Oil switches should be examined once a week to insure that the contacts have not become burnt.

The oil in the oil switch tanks should be renewed every three or six months, depending upon the number of times the switch has been opened under load.

Spares for switches should always be kept at hand.

Transformer temperatures should be taken daily; the transformer temperatures should never be permitted to exceed 70 degrees C. If the temperature approaches 70 degrees C., increase the amount of ventilation of the transformer room or reduce the load on the furnace.

Transformer oil gauges should be frequently inspected to see that the oil shows in the glass.

A sample of oil should be taken from the bottom of each transformer every six months and tested for moisture. At the end of each year all the oil in the transformers should be removed and either filtered or renewed.

ADDITIONS.

In producing plain carbon electric steel the additions used are preferably:

50 ferro-silicon.
80 ferro-manganese.

In producing alloy steels:

For tungsten steel,

Tungsten powder, 97 to 99 per cent tungsten, or

Ferro-tungsten, 70 to 75 per cent tungsten.

For chrome steel,

Ferro-chrome, 66 to 70 per cent chrome.

For vanadium steel,

Ferro-vanadium, 45 per cent vanadium.

As illustrating the necessity of forming and removing the first slag in electric furnace operation, the following data is given:

Heat No. 56. July 17, 1919.

First Slag.

Sulphur (including sulphides)	4.22
Manganese	.42
Phosphates (as P_2O_5)	4.47
SiO_2	16.70
Iron (and Al_2O_3)	7.00
Lime (CaO)	58.50
Magnesia (MgO)	9.35

Steel.

Carbon	.23
Manganese	.67
Silicon	.25
Phosphorous	.014
Sulphur	.021

Heat No. 65. July 22, 1919.

(In this heat the first slag was not removed, but was held throughout the heat.)

Final Slag.

Phosphates (as P_2O_5)	.25
SiO_2	38.50
Iron (and Al_2O_3)	35.75
Lime (CaO)	12.98
Magnesia (MgO)	12.50

Steel.

Carbon	.18
Manganese	.68
Silicon	.27
Phosphorous	.045
Sulphur	.027

These analyses show that failing to remove the first slag increases the phosphorous of the steel from .014 to .045; furthermore, the increase of iron in the slag from 7.00 per cent to 35.75 per cent shows steel going into the slag which should go into the ladle.

The following analyses show the extent to which sulphur is taken up in the second slag and how the carbon and the manganese content are increased by the additions:

Heat No 84. Aug. 14, 1919.

	Before Formation of Second Slag.	*Steel in Ladle.*
Carbon	.048	.32
Manganese	.09	.66
Silicon	.22	.15
Phosphorous	.02	.025
Sulphur	.045	.017

The following chemical and physical results, selected at random, were obtained from billets forged from ingots:

Heat	C.	Si.	Mn.	S.	P.	Elastic Limit	Tensile Strength	Elongation in 2"	Reduction of Area	Cold Bend 1"
								Per Cent	Per Cent	Degrees
24	.248	.23	.64	.014	.027	36,400	64,550	34	28	180
30	.27	.14	.64	.031	.018	35,000	62,750	27.5	37	180
37	.39	.19	.65	.024	.019	46,500	78,100	23	35	180
46	.29	.20	.70	.017	.027	47,950	74,750	33	34	180
50	.39	.17	.65	.019	.023	41,800	79,850	26	22	180
57	.38	.24	.76	.022	.029	46,150	95,250	23	16	180
66	.28	.29	.63	.023	.029	33,200	84,150	28	25	180

Photo 8 shows one cold bend test specimen.

After considerable experimenting, it was found that the crop end of a 22-inch ingot could be cut off by the oxyhydrogen torch for $5.00 (labor plus material) and that the cropped ingot weighing about 5,000 pounds could be turned into store at $0.08 per pound, which included all costs of production. When it is considered that the Navy Department paid as high as $0.59 per pound for nickel steel forging ingots during the war, the possibilities of electric furnace production at navy yards are interesting.

Photo 9 shows ingots and castings produced by the electric steel furnace.

To reduce to a minimum the loss of time resulting from an electrode breaking through the threaded joint, a number of electrodes were kept on hand made up complete (Photo 10). A broken electrode could be removed and a new one inserted in comparatively few minutes.

Photo 11 shows the absence of scull in the ladle after pouring. The scull in the ladle from a six-ton heat usually averaged 200 pounds.

The following data obtained from heats 1 to 50 are of value. The ton of metallic charge has been adopted as the basis of calculation and comparison. The metallic charge consists of all ferrous material (including additions) used in the charge and exclusive of slag-making materials such as lime, fluor spar and sand.

NOMINAL AVERAGES AND ACTUAL AVERAGES.

In order to compare the performance of the furnace for definite periods of time, the operation was divided into periods of 10 heats. This eliminated the major fluctuations of individual heats; for instance, where castings were made in one heat, but, through error, credited to a subsequent heat. Five complete sets of figures covering the first fifty heats were obtained. The average values of these five sets are the *nominal*

averages. The *actual* averages are the averages of the entire fifty heats, the discrepancy between the nominal averages and the actual averages is due to the fact that all heats were not of the same size and consequently the ratio of the various percentages was not uniform. The nominal averages were of value in following the variations in furnace performance during the progress of the work. The actual averages could not be obtained until the conclusion of the fifty heats. The actual averages are of more value than the nominal averages.

Table A gives the charging materials in detail for heats 41-50.

Table B gives the percentages of the charging materials for heats 1-50 in groups of 10 heats.

Table C gives a summary of the charges for heats 1-50.

Table D gives the total charge, the product, and the furnace losses for heats 41-50.

Table E gives the output in pounds for heats 1-50 in groups of ten heats.

Table F gives the output in percentages for heats 1-50 in groups of ten heats.

In connection with Table D, percentages are figured in three different ways, A, B, and C as follows:

A. On the basis of the total Metallic Charge, taken as 100 per cent:

1. As to Input:

(*a*) Primary Charge, per cent.....	97.75
(*b*) Added Metallic Charge, per cent.....	2.25
(*c*) Total Metallic Charge, per cent.....	100.00

2. As to Output:

(*d*) Good Castings, per cent.....	31.12
(*e*) Ingots, per cent.....	47.00

(*f*) Total Good Castings and Ingots, per cent	78.12
(*g*) Defective Castings, per cent	.52
(*h*) Returns, per cent	17.51
(*i*) Total Output Product, per cent	96.15
(*j*) Foundry and Furnace Losses	3.85
(*k*) Total Metallic Charge, per cent	100.00

B. On the Basis of the Furnace Product:

1. Output only taken as 100 per cent:

(*a*) Good Castings, per cent	32.40
(*b*) Ingots, per cent	48.85
(*c*) Total Good Castings and Ingots, per cent	81.25
(*d*) Defective Castings, per cent	.54
(*e*) Returns, per cent	18.21
(*f*) Total Furnace Product, per cent	100.00

C. On the Basis of Castings and Ingots:

1. Output only taken as 100 per cent:

(*a*) Good Castings, per cent	39.63
(*b*) Ingots, per cent	59.70
(*c*) Good Castings and Ingots, per cent	99.33
(*d*) Defective Castings, per cent	.67
(*e*) Total Castings and Ingots, per cent	100.00

Note: The above data is taken from Group of Heats 41 to 50.

The "A" Basis of Comparison is useful in comparing the Input and the various Output subdivisions with one another. In the earlier heats the "furnace and foundry losses" were found to be abnormally high. A stricter accounting of the

material was made with the latter heats, so that these losses appear considerably smaller in them. This loss appears very small in all heats where ingots only were made, which indicates that the bulk of the losses should be charged to the castings and not to the ingots alone.

The "B" Basis of Comparison is useful in comparing the distribution of the product from the furnace as showing the relative portions going to castings, to ingots, to defective castings, and to returns. The "B" system is useful in following the practical work of the foundry alone, whereas the "A" system applies more strictly to the furnace operation.

The "C" Basis of Comparison is useful in showing the percentage of useful castings, defective castings and ingots.

The "A" Basis of Comparison is based on the Total Metallic Charge as 100 per cent. The "B" Basis of Comparison is based on the Total Product of the Furnace (*i.e.*, the metal poured into the ladle) taken as 100 per cent. The "C" Basis of Comparison is based on the sum or total of the ingots of castings—both good and defective.

COSTS.

The costs given in Tables G to L are all based on the ton of metallic charge.

Table G gives the cost of charging materials.

Table H gives the cost of electrodes.

Table I gives the cost of power.

Table J gives the cost of labor.

Table K gives the cost of refractories used in repairing the hearth, roof and walls, based on data covering 80 heats.

Table L gives a summary of the costs per ton of metallic charge.

SUMMARY.

From the data contained in Tables A to L, the following results and costs are obtained:

During the first 50 heats there were actually produced (Table E):

	Pounds.	*Per cent.*
Good Castings and Ingots	443,553	70.66
Miscellaneous Losses and Returns	184,216	29.34
Total of Metallic Charges	627,769	100.00

The nominal average for good castings and ingots found from the averages of percentages was 69.82

The maximum being (No. 41-50) 78.12

The minimum being (No. 1-10) 54.40

Showing the improvement in the percentage of usable or salable material as experience was gained.

The corresponding losses and returns were:

(For nominal average), per cent 30.18

(For Heat No. 41-50) 21.88

(For Heat No. 1-10) 45.60

It will be noted that the actual per cent of Miscellaneous Losses and Returns was 29.34

The actual per cent of Foundry Scrap used or returned was 26.16

Leaving the total Furnace Loss as 3.18

Properly speaking, a small fraction of this total furnace loss should be charged to the ladle and possibly another fraction to the foundry losses.

The total cost of materials and labor for each metallic ton of charge (Table L) $40.6218

Actual percentage of good castings and ingots, per cent 70.66

Total cost per Ton of Good Castings and Ingots ($40.6218/0.7066) $57.4891

Cost per Pound of Good Casting material and Ingots ..... $ 0.02875

From the total furnace loss of 3.18 per cent, if we assume the efficiency of the furnace as 96.82 per cent, then the cost of Steel placed in the ladle is ($40.6218/0.9682) $41.9560

Cost per Pound in the Ladle ..... $ 0.020978

Or in round numbers ..... $ 0.021

Assuming an overhead charge of 100 per cent, the total cost in the ladle will be, per pound ..... $ 0.042

The cost of an ingot can be considered slightly higher as the ladle and handling charges should be added (assumed as .003 per pound) ..... $ 0.045

In case the ingot had 30 per cent of the top removed as discard, the cost, exclusive of the labor necessary, would be about $0.045/.70 cost per pound ..... $ 0.0643

The labor and material necessary for cropping the end is .001 per pound ..... .001

Cost of forging Ingots f.o.b. foundry ..... .065

Based on these figures, cropped ingots were delivered to Supply Officer, Mare Island, at $0.08 per pound, as stated on page 295.

Photo 12 shows a heat of electric steel being tapped into the ladle.

Рното. 1.

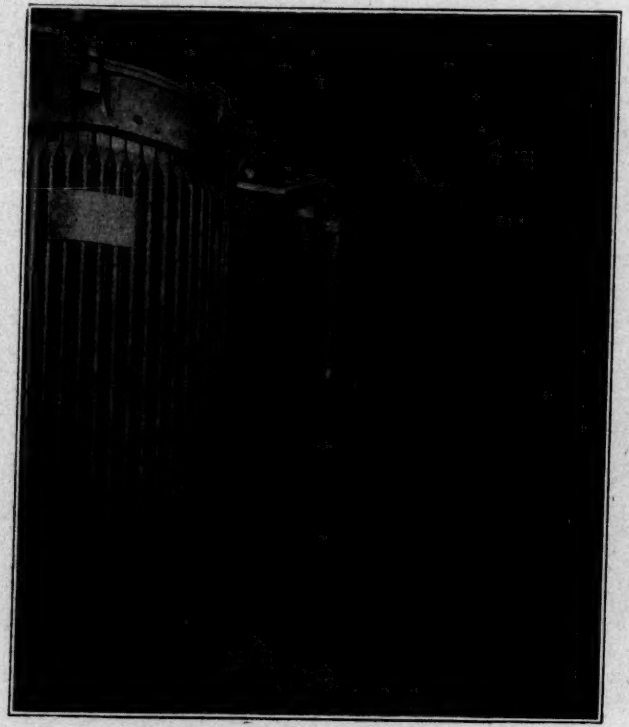


Photo. 2.

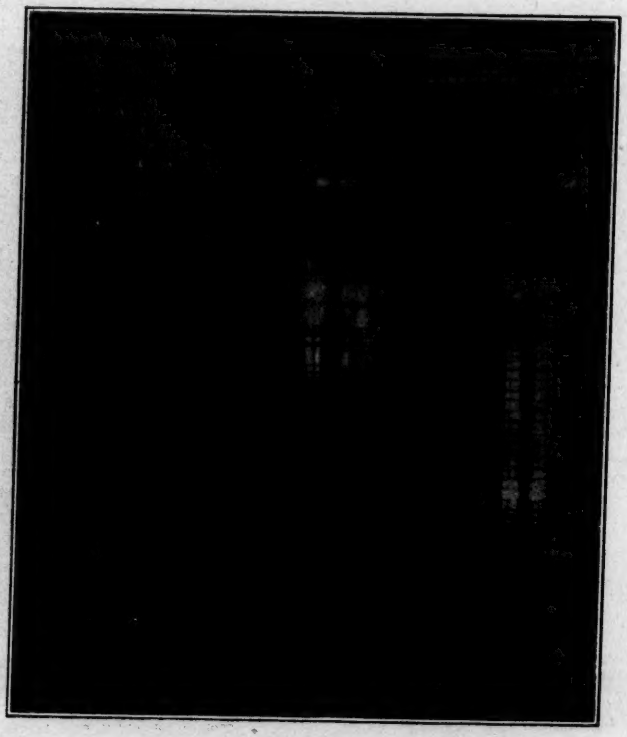
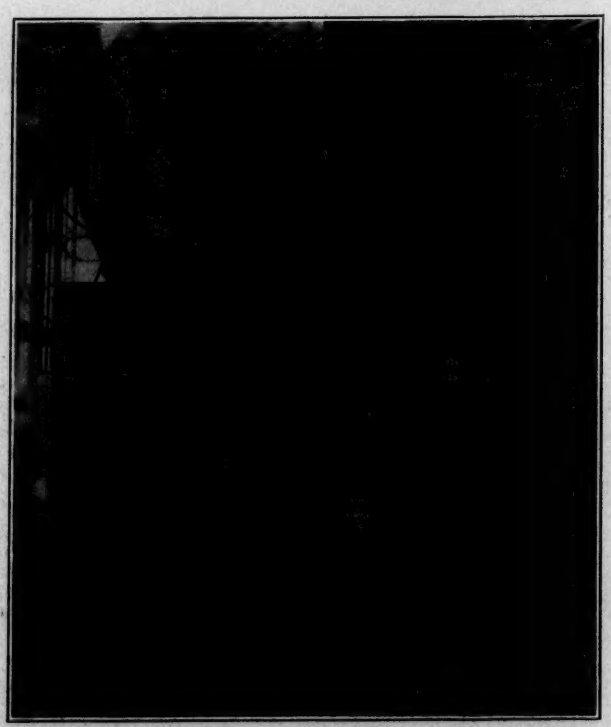


Photo. 3.

Photo. 2.

Photo. 3.



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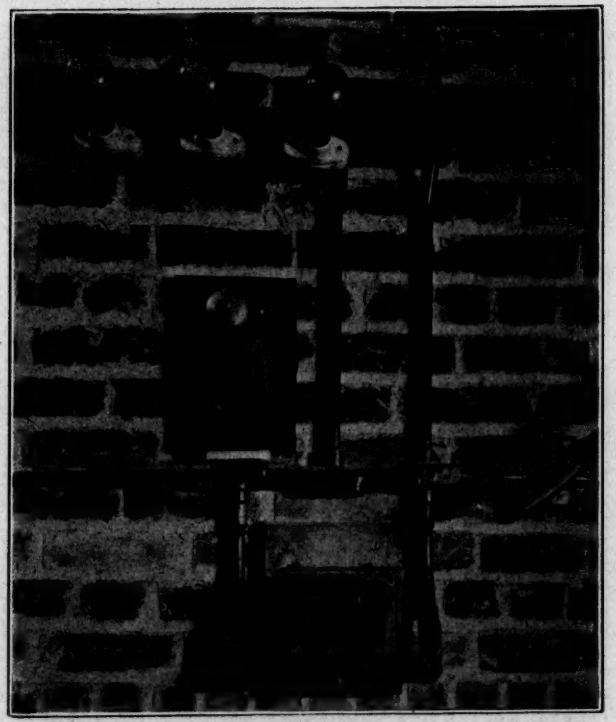


Photo. 5.

Photo. 6.

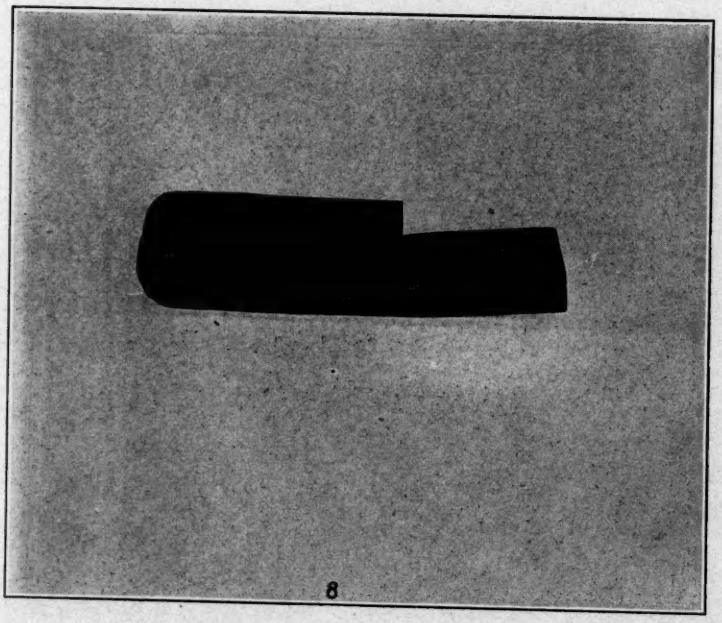


Photo. 8.

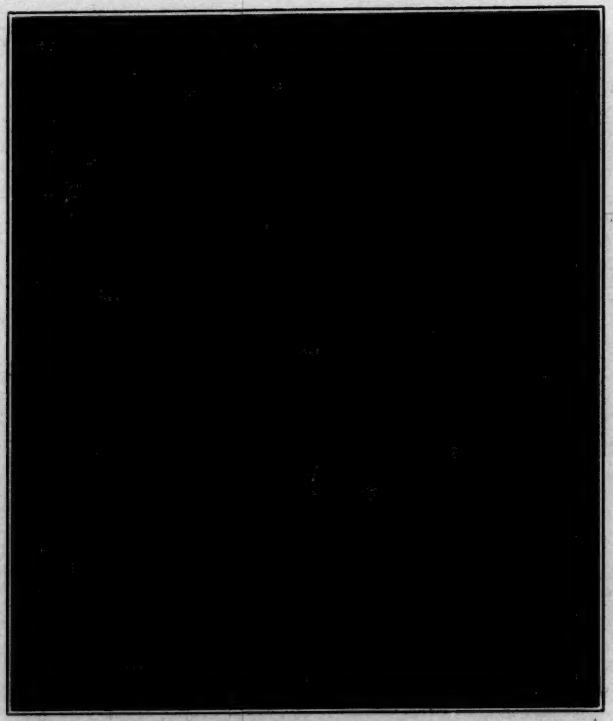


Photo. 7.

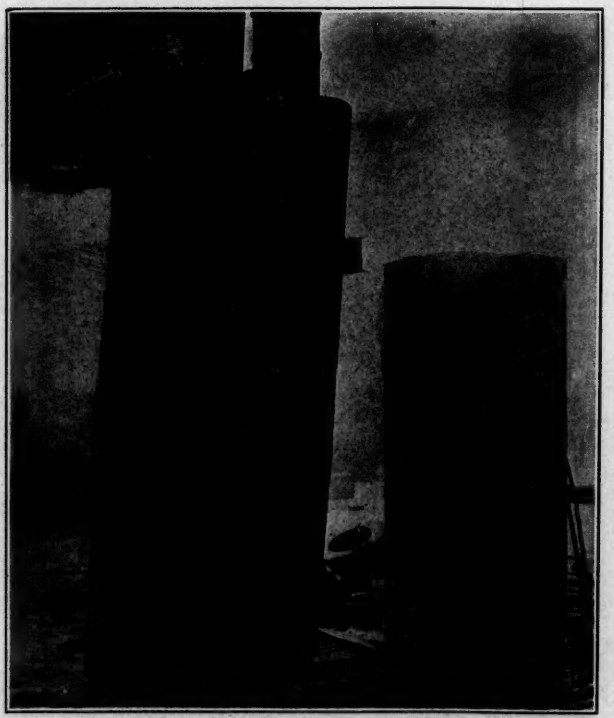


Photo. 10.

Photo. 9.

Photo. 11.

Photo. 12.

TABLE A.

HEAT NOS. / MATERIAL	#41	#42	#43	#44	#45	#46	#47	#48	#49	#50	#41 - #50 Pounds	%	%
Foundry scrap	4065		4500	3580	3254	2545	2560	4065	4600	4030	33197	25.94	26.55
Plate "	3010	5040	4610	5490	5650	6490	6460	5002	6390	4995	53137	41.56	42.55
Trimming "	1976	3005	905	690	2018	2010	2030	2025	2050	1550	18259	14.26	14.58
Punching "	3052	4030	2070	2330	1096	1585	1010	1015	2946	1500	20433	15.98	16.32
Shell "													
PRIMARY CHARGE	12121	12075	12085	12060	12028	12430	12060	12107	15985	12055	125006	97.74	100.00
Pig Iron		250	300					600	135	240	1525	1.19	
Scale													
Fe Si	60	50	50	50	55	50	50	60	55	50	530	.42	
Fe Mn	95	65	85	88	65	65	65	75	92	90	785	.62	
Fe Cr													
Nickel													
Al	3.25	3.25	3	3.5	3.5	3.25	3.5	3.5	5	3.5	35.25	.03	
ADDED MET. CH.	158.25	368.25	438	141.5	123.5	118.25	118.5	738.5	287	383.5	2875.25	2.26	
TOTAL MET. CH.	12279.25	12443.25	12523	12201.5	12151.5	12548.25	12178.5	12845.5	16272	12438.5	127881.25	100.00	lbs/ton
Lime	300	300	300	350	350	300	300	350	350	350	3250	2.54	51.30
Fluor Spar	60	60	60	60	60	60	60	60	60	60	600	.47	9.47
Coke	25	25	30	25	25	25	25	35	35	50	300	.24	4.73
Sand													
NON MET. CH.	385	385	390	435	435	385	385	445	445	460	4150	3.25	65.50
GRAND TOTAL	12764.25	12828.25	12913	12636.5	12586.5	12933.25	12563.5	13290.5	16717	12898.5	132031.25	103.25	

TABLE B.

CHARACTER OF CHARGE IN PERCENTAGES.

HEAT NOS.	1-10	11-20.	21 - 30	31 - 40	41 - 50	NOMINAL AVERAGE
MET.CHARGE TONS	49.269	68.367	65.350	66.937	63.941	62.773
	%	%	%	%	%	%
Foundry Scrap	9.38	26.33	35.95	28.96	25.94	25.312
Plate "	18.82	29.63	42.50	29.70	41.56	32.442
Trimming "	26.53	18.78	10.10	10.75	14.26	16.084
Punching "	27.09	12.03	7.95	25.90	15.98	17.790
Shell "	12.93	9.14				4.414
PRIMARY CHARGE	94.75	95.91	96.50	95.31	97.74	96.042
Pig Iron	2.68	1.85	.57	2.69	1.19	1.796
Scale	1.42	1.28	1.95	0.90		1.110
Fe Si	.42	.39	.40	.40	.42	.406
Fe Mn	.70	.54	.55	.67	.62	.616
Fe Cr						
Nickel						
Aluminum	.03	.03	.03	.03	.03	.030
ADDED MET. CH.	5.25	4.09	3.50	3.69	2.26	3.958
TOTAL MET. CH.	100.00	100.00	100.00	100.00	100.00	100.000
Lime	3.80	3.79	3.79	2.35	2.54	3.254
Fluor Spar	.78	1.12	.99	.45	.47	.762
Coke	.43	.36	.38	.25	.24	.332
Sand	.27	.30	.31	.05		.188
NON MET. CH.	5.28	5.57	5.47	3.11	3.25	4.536
GRAND TOTAL	105.28	105.57	105.47	103.11	103.25	104.536

TABLE C.

SUMMARY OF CHARGES FOR FIFTY HEATS

	TOTAL LBS FOR 50 HEATS	AVERAGE LBS PER HEAT	ACTUAL %	TOTAL %
Foundry Scrap	164210	3284.20	26.16	
Plate "	207596	4151.92	33.08	
Trimming "	97611	1952.22	15.55	
Punching "	108584	2171.68	17.30	
Shell "	25216	504.32	4.00	
PRIMARY CHARGE	603217	12064.34		96.09
Pig Iron	11039	220.78	1.76	
Scale	6900	138.00	1.10	
Fe Si	2548	50.96	.41	
Fe Mn	3838	76.76	.61	
Fe Cr				
Nickel				
Al	185	3.70	.03	
ADDED MET. CH.	24510	490.20		3.91
TOTAL MET. CH.	627727	13554.54		100.00
Lime	20280	405.60	3.23	
Fluor Spar	4810	96.20	.78	
Coke	2040	40.80	.32	
Sand	1150	23.00	.18	
NON MET. CHARGE	28280	565.60		4.51
GRAND TOTAL	656007	13120.14		104.51

TABLE D.

HEAT NO.	DATE	PRIMARY CHARGE LBS	ADDED METALLIC CHARGE	TOTAL CHARGE LBS	TOTAL CHARGE TONS	GOOD CASTINGS LBS	INGOTS LBS	INGOTS & CASTINGS LBS	DEFECTIVE CASTINGS LBS	RETURNS LBS	TOTAL PRODUCT LBS	FURNACE & FOUNDRY LOSSES LBS	LOSSES %
41	June 11	12121	158	12279	6.139	6850		6850		3315	12165	114	.93
42	" 12	12075	368	12443	6.222	4223	7340	11563		1070	12633	- 190	-1.52
43	" 13	12085	438	12523	6.261	2568	7548	10116		2450	12566	- 43	-0.34
44	" 14	12060	142	12202	6.101	2158	7570	9728	90	1464	11282	920	7.54
45	" 16	12028	123	12151	6.076	2120	7550	9670		1730	11400	751	6.18
46	" 17	12430	118	12548	6.274	1946	7508	9454	231	2460	12165	383	3.06
47	" 18	12060	119	12179	6.089	2362	7248	9610		2420	12030	149	1.23
48	" 19	12107	739	12846	6.423	2005	7680	9683		1880	11563	1283	10.00
49	" 20	15985	287	16272	8.136	12155		12166	82	3120	15368	904	5.55
50	" 23	12055	384	12439	6.220	1421	7605	9026	266	2500	11792	647	5.20
TOTALS		125006	2876	127882	63.941	39817	60049	99866	669	22429	122964	491 8	
A %		97.75	2.25	100.00		31.12	47.00	78.12	.52	17.51	96.15	3.85	
B %						32.40	46.85	81.25	.54	18.21	100.00		
C %						39.68	59.70	99.38	.67				

TABLE E.
SUMMARY FOR FIFTY HEATS
OUTPUT IN POUNDS

	1 - 10	11 - 20	21 - 30	31 - 40	41 - 50	1 - 50	AVERAGE TONS
	LBS	LBS	LBS	LBS	LBS	LBS	PER HEAT
GOOD CASTINGS	42853	27501	44514	28706	39617	183071	1.83071
INGOTS	11076	76014	40820	75125	60049	260468	2.60468
INGOTS & CASTINGS	53609	103515	84734	101829	99666	443855	4.43553
DEFECTIVE CASTINGS	2596	1418	3017	1761	699	9507	.09477
RETURNS	21142	12956	22192	16250	22429	94971	.94971
TOTAL PRODUCT	77349	117885	109943	119860	122994	548031	5.48001
FURNACE & FOUNDRY LOSSES	21189	18842	20768	14015	4918	79732	.79736
TOTAL CHARGE	98538	136728	130711	133875	127912	627769	6.27739

TABLE F.

SUMMARY OF OUTPUTS FOR FIFTY HEATS
IN PER CENT OF CHARGE.

	1 - 10	11 - 20	21 - 30	31 - 40	41 - 50	NORMAL AVERAGE
TOTAL CH. IN LBS	98538	136733	130711	133875	127882	12554.6
OUTPUT	%	%	%	%	%	%
GOOD CASTINGS	43.15	20.1	34.07	21.45	31.12	29.978
INGOTS	11.25	55.60	30.76	54.60	47.00	39.842
INGOTS & CAST'S	54.40	75.70	64.83	76.05	78.12	69.820
DEFECTIVE CAST'S	2.64	1.03	2.31	1.33	.52	1.566
RETURNS	21.47	9.47	16.96	12.15	17.51	15.514
TOTAL PRODUCT	78.51	86.20	84.10	89.54	96.15	86.900
FURNACE & FOUNDRY losses	21.49	13.80	15.90	10.46	3.85	13.100
TOTAL METALLIC CHARGE	100.00	100.00	100.00	100.00	100.00	100.000

TABLE G.

COST OF MATERIAL PER TON OF METALLIC CHARGE

Based on actual weights used during Heats NO.1 to 50, Incl.

MATERIALS	Percent %	Price per Ton $	Cost $	TOTAL COST $
Foundry Scrap	26.16	00.00		
Plate Scrap	33.08	10.00	3.308	
Turnings	15.55	10.00	1.555	
Punchings	17.30	10.00	1.730	
Shells	4.00	10.00	.400	
Primary Charge	96.09			6.993
Pig Iron	1.76	47.30	.832	
Scale	1.10	5.00	.055	
Ferro Silicon	.41	240.00	.984	
Ferro Manganese	.61	204.00	1.244	
Ferro Chromium				
Nickel				
Aluminum	.03	666.66	.200	
Metallic Additions	3.91			3.315
Total Metallic Charge	100.00			10.308
Lime	3.23	29.00	.937	
Fluor Spar	.78	55.00	.429	
Coke	.32	5.50	.018	
Sand	.18	1.50	.003	
Non-metallic Charge	4.51			1.387
Grand Totals	104.51			11.695

TABLE H.

ELECTRODE COSTS

POUNDS Used as Taken from Log Sheet

HEAT No.	#1 Dr.	#1 Cr.	#2 Dr.	#2 Cr.	#3 Dr.	#3 Cr.	#4 Dr.	#4 Cr.
# 1-10	1735		2275		1705		2275	
11-20	610		560	230	585	475	585	
21-30	585		1755	440	1195	250	1220	
31-40	585	250			685		585	
41-50	585		1170		585		585	
TOTALS	4100	250	5760	670	4755	725	5250	000
NET	3850		5090		4030		5250	
¢	21.12		27.94		22.11		28.83	

Total Quantity charged to the Electrodes	18,220 lbs.
Electrodes in position, broken ends, etc.	5,383
Net Quantity used	12,837 lbs.
Amount per Ton Metallic Charge 12,837 / 313.87	40.9 lbs.

Cost of Electrodes per Ton of Metallic Charge

$0.0875 x 40.9 = $3.5788

TABLE I.

COST OF ELECTRIC POWER

Taken from Log Data for Heats No.1-50 Inc.

HEAT NOS.	K.W.-HRS. PRELIMINARY HEATING	K.W.-HRS. MELTING & REFINING	CHARGE IN TONS	K.W.-HRS. per TON
1 to 10	15,000	59,000	49.269	1198
11 to 20	200	66,900	68.367	979
21 to 30	700	67,200	65.355	1028
31 to 40	3,600	64,000	66.938	956
41 to 50		63,300	63.941	990
TOTALS	19,500	320,400	315.870	
	62.2	1,020	1.000	

	KW.-HRS.
Direct Current Power used by Auxiliaries, per heat	50
Direct Current Power per Ton $\frac{50 \times 50}{315.87}$	8

Rate for Power $ 0.0125 per K.W.-HR.

For Melting and Refining $ 0.0125 x	1,020	equals	$ 12.750
For Preliminary Heating $ 0.0125 x	62.2	equals	$ 0.777
For Auxiliaries $ 0.0125 x	8	equals	$ 0.100
TOTAL COST PER TON FOR POWER			$ 13.627

TABLE K.

COST OF REFRACTORIES
(Roof and Walls)
Taken from Log Data.

Heat No.	QUANTITY	MATERIAL	UNIT PRICE	AMOUNT
22	100	W.Star Silica 9" Straights	.2082	20.82
	50	Magnesite 9" Straights	.8010	40.05
	23	W.Star Silica #2 Arch	.2082	4.7886
	6	Metalbase #1 Key	.751	45.06
	300 Lbs.	Canister	.012	3.60
		Total for Heat #22		$114.3186
31	365	W.Star Silica 9" Straights	.2082	75.993
	56	W.Star Silica #2 Wedge	.2082	11.6592
	49	Magnesite Straights	.8010	39.249
	33	Metalbase 15-1/2" Jambs	1.1000	36.300
	15	Metalbase 9" Jambs	.8510	12.765
	91	Franklin Crown	.36935	33.6109
	500 Lbs.	Magnesite	.0625	31.25
	25 Lbs.	Pitch	.033	.826
				241.6551
63	350	W.Star Silica	.2082	72.870
	24	W.Star Silica #2 Wedge	.2082	4.9968
	30	Magnesite	.801	24.0300
	1	24" Firebrick slab	3.00	3.000
	330 Lbs.	Magnesite	.067	22.11
		Total for Heat #63		127.0068
77	416	W.Star Silica #1 End Wedge	.2082	86.6112
	44	W.Star Silica #2 Side Arch	.2082	9.1608
	88	W.Star Silica #5 Side Arch	.2082	18.3216
	50	W.Star Silica 6 x 9 #1 Wedge	.2960	14.800
	10	W.Star Silica 6 x 9 Straight	.5170	5.170
	925	W.Star Silica 9" Straights	.2082	192.585
		Total for ROOF Heat #77		324.6486
	600 Lbs.	Magnesite	.067	40.20
	43	Magnesia Brick 9" Straights	.8010	34.443
	35	Magnesia Brick #2 Wedge	.8010	28.035
	11	Chrome Brick 9" Straights	.901	9.911

		Total for ROOF Heat #77		324.6486
600	Lbs.	Magnesite	.067	40.20
45		Magnesia Brick 9" Straights	.8010	36.445
35		Magnesia Brick #2 Wedge	.8010	28.035
11		Chrome Brick 9" Straights	.901	9.911
10		Metalkase 13-1/2" Straight	1.100	11.000
14		W.Star Silica #3 Arch	.2082	2.9148
		Total for Walls Heat #77		86.3038
81	50	W.Star Silica 9" Straights	.2082	10.41
	1	Fire Clay Slab 12" x 24"		5.00
		Total for Heat #81		15.41

COST OF REFRACTORIES
(Bottom)

HEAT NOS.	MAGNESITE	MAGNITE	TOTALS
1 to 10	400		
11 to 20	2450		
21 to 30	2625		
31 to 40	1000		
41 to 50	700	1400	
TOTALS	7175	1400	8575

Price of Magnesite per 2000 lb. ton $35.00
$35.00 x (8575/2000) 150.06
Cost per ton of Metallic Charge $\frac{\$150.06}{313.67}$ 0.4844

SUMMARY

HEAT No.				WALLS	ROOF
22	Brick,	Misc.	types	114.3186	
31	"	"	"	241.6531	
65	"	"	"	127.0068	
77	"	"	"	86.3038	324.6486

		Total for Heat #[illegible]		$114.3186
31	365	W.Star Silica 9" Straights	.2082	75.993
	56	W.Star Silica #2 Wedge	.2082	11.6592
	49	Magnesite Straights	.8010	39.249
	33	Metalkase 13-1/2" Jambs	1.1000	36.300
	15	Metalkase 9" Jambs	.8510	12.765
	91	Franklin Crown	.36935	33.6109
	500 Lbs.	Magnesite	.0625	31.25
	25 Lbs.	Pitch	.033	.826
				241.6531
63	350	W.Star Silica	.2082	72.870
	24	W.Star Silica #2 Wedge	.2082	4.9968
	30	Magnesite	.801	24.0300
	1	24" Firebrick slab	3.00	3.000
	330 Lbs.	Magnesite	.067	22.11
		Total for Heat #63		127.0068
77	416	W.Star Silica #1 End Wedge	.2082	86.6112
	44	W.Star Silica #2 Side Arch	.2082	9.1608
	88	W.Star Silica #3 Side Arch	.2082	18.3216
	50	W.Star Silica 6 x 9 #1 Wedge	.2960	14.800
	10	W.Star Silica 6 x 9 Straight	.3170	3.170
	925	W.Star Silica 9" Straights	.2082	192.585
		Total for ROOF Heat #77		324.6486
	600 Lbs.	Magnesite	.067	40.20
	43	Magnesia Brick 9" Straights	.8010	34.443
	35	Magnesia Brick #2 Wedge	.8010	28.035
	11	Chrome Brick 9" Straights	.901	9.911
	10	Metalkase 13-1/2" Straight	1.100	11.000
	14	W.Star Silica #3 Arch	.2082	2.9148
		Total for Walls Heat #77		86.3038
81	50	W.Star Silica 9" Straights	.2082	10.41
	1	Fire Clay Slab 12" x 24"		3.00
		Total for Heat #81		13.41

COST OF REFRACTORIES
(Bottom)

HEAT NOS.	MAGNESITE	MAGNITE	TOTALS

(Bottom)

HEAT NOS.	MAGNESITE	MAGNITE	TOTALS
1 to 10	400		
11 to 20	2450		
21 to 30	2625		
31 to 40	1000		
41 to 50	700	1400	
TOTALS	7175	1400	8575

Price of Magnesite per 2000 lb. ton $35.00

$35.00 x (8575/2000) 150.06

Cost per ton of Metallic Charge $\frac{\$150.06}{313.87}$ 0.4844

SUMMARY

HEAT No.		WALLS	ROOF
22	Brick, Misc. types	114.3186	
31	" " "	241.6531	
63	" " "	127.0068	
77	" " "	86.3038	324.6486
81	" " "	13.4100	
	COST, TOTAL	582.5923	324.6486
	No of HEATS	80	76
	COST PER HEAT	7.2712	4.2717
	Cost Per Ton of Metallic Charge of 6.2774 tons	1.1422	0.6805

Note:- Costs of materials as given on Bu. Cont. 39391.

TABLE J.

COST OF LABOR

OPERATION				
			Per diem	
	2	Melters	at $8.08	$16.16
	1	Melter	at 7.56	7.56
	1	Electrician	at 6.40 (20% time)	1.28
	5	Helpers	at 4.64	23.20
		Daily Cost of Labor		$48.20
	$\frac{\$48.20 \times 50}{315.87}$		:: Cost per Ton	$ 7.6783
MAINTENANCE				
	Assumed at 20 % of the above			
	$7.6783 x 0.20		equals	$ 1.5356

TABLE L.

TOTAL COST (PER TON OF METALLIC CHARGE)

OPERATION:-			
Table Q	Primary Charge	$6.9930	17.20
	Metallic Additions	3.3150	8.16
	Non-Metallic Additions	1.3870	3.41
	Total Charge $11.695, 28.77%		
Table H -	Electrodes	3.5788	8.82
Table I-	Power	13.6270	33.56
Table J-	Labor	7.6783	18.80
	Total for Operation	36.5791	90.05
MAINTENANCE			
Table K-	Refractories:- Bottom	.4844	1.19
	Roof	.6805	1.67
	Walls, etc	1.1422	2.81
	Tools, oil, etc (Estimated)	.2000	0.49
	Labor (Estimated)	1.5356	3.79
	Total for Maintenance	4.0427	9.95
	TOTAL COST PER METALLIC TON	$ 40.6218	100.00
	TOTAL COST PER METALLIC LB.	$ 0.02031	

TRIALS OF NEW FIFTY-FOOT STEAMERS EQUIPPED WITH TURBINE AND REDUCTION GEAR.

By Lieutenant-Commander William C. Owen, U. S. Navy, Member.

GENERAL DESCRIPTION.

The launch is a twin-screw vessel, driven by two high-speed Poole Turbines through planetary reduction gears, and is designed for a speed of 800 R.P.M. of the propeller, at about 20 tons trial displacement, with each turbine developing 100 shaft horsepower.

GENERAL DATA.

Length over all, feet	50
Beam, feet and inches	9–10¼
Weight, without crew, pounds	44,700
Draft to bottom of keel, feet and inches	3–8
Fuel oil capacity, gallons	300
Fresh water capacity, gallons	186
Cruising radius at full speed, miles	60.25
Cruising radius at 600 R.P.M., miles	74.5

The launch is divided into four compartments, as follows: Forward: crew quarters, fire-room, engine-room; and cock-pit for passengers aft. The cock-pit is fitted for carrying approximately twenty passengers, and the launch has a carrying capacity of approximately fifty passengers and a crew of five men.

MACHINERY.

The main engines are Poole high-speed turbines of the compound impulse velocity stage type, with a working pressure of 250 pounds per square inch gauge.

The turbines are designed to develop 100 shaft horsepower each, at 8,000 revolutions, and are geared down in the ratio of ten to one. The photograph here shown, marked Figure 1, gives a good reproduction of the turbine rotor, reduction gear, forced lubrication pump and thrust block.

Aluminum has been used in the construction of these engines, in the casings and other parts where practicable, to reduce weight. Each turbine and reduction gear weighs approximately 1,400 pounds.

TURBINES.

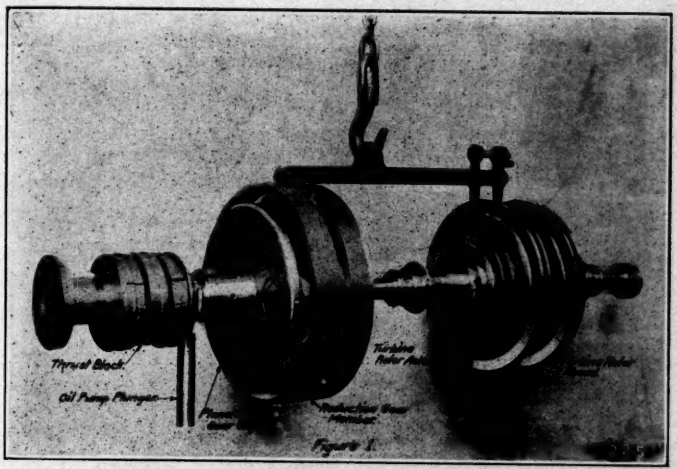
There are two pressure stages in the ahead turbines; both are velocity compounded. Incorporated in the same casing and in the after end, is the astern turbine, which consists of one pressure stage, velocity compounded. It has about 55 per cent of the power of the ahead turbine.

The rotor wheels are secured to tapered sleeves, shrunk on the shaft. The sleeves, one for each wheel, are shrunk and keyed to the shaft and secured by three keeper bolts. The inside of the wheels are machined to a machine-fit with the sleeves and is held in place by a key extending through the sleeve to the shaft, and by securing rings which screw on to the tapered sleeve on the shaft. The securing rings are secured in place by keeper bolts which project through the rings into the wheel, thereby preventing the wheel from slipping.

The two wheels in the ahead turbine are of the same size, and are 16 inches in diameter at the center of the blades. The astern wheel is 10 inches in diameter at the center of the blades. Each wheel carries two rows of blades.

The shaft is supported by self-aligning bearings of the spherical-seat type. The forward bearing is located in the forward end of the turbine casing, and the after bearing in the forward end of the reduction-gear casing.

The central part of the shaft is 23⅛ inches long and 3½ inches in diameter, which carries the rotor wheels; and forward and aft of rotor wheels the shaft revolves in a series



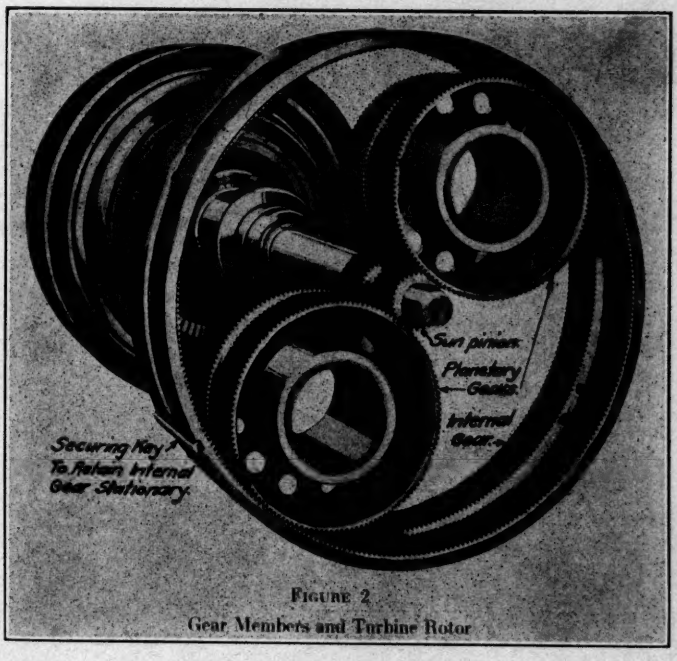


FIGURE 2
Gear Members and Turbine Rotor

of radial-slotted projections on the turbine case forming part of the shaft gland packing. The shaft gland packing is completed by a securing ring on each end of the shaft, where the diameter is reduced and the ring revolves in radial-slotted projections on the turbine case. Sealing steam from the auxiliary exhaust line is admitted to an annular space at the center of the packing.

The sun pinion gear is cut on the after end of turbine shaft, and consists of a right and left hand helix.

The periphery of the rotor wheels is ⊥ slotted. The roots of the rotor blades are of a corresponding cross-sectional shape. A starting place for inserting the roots of the blades is provided, and after all blades are inserted in the rotor slot, a locking-piece is inserted and welded into place.

The distance between the blades is maintained by the shape of their roots. After all the blades are assembled on a rotor, a shroud-ring in sections is fitted over the tips of the blades.

The diaphragms which carry the stationary blades are made in halves and are set in slots in the upper and lower halves of the casing and are secured with stud bolts. The diaphragms can be removed when the upper casing is lifted, by coming up on the stud bolts.

The diaphragm which carries the steam nozzles of the second stage is made in halves also, and set in slots in the upper and lower half of the casing. The lower diaphragm can be rolled out when the upper half of the casing is lifted. The diaphragms are fitted with radial packing to prevent leakage from the first to the second stage. There are eight nozzles in the diaphragm, all of which are in the lower half.

There is one nozzle bolted to the lower part of the forward turbine casing for delivering steam to the ahead turbine, and two nozzles bolted to lower part of the after turbine casing for admitting steam to the astern turbine. Steam admitted is controlled directly by the maneuvering valve. A strainer and stop-valve are fitted in the main steam line ahead of the maneuvering valve, and there is only one strainer and one stop-

valve to supply the two maneuvering valves. A hand trip gear is installed so that the stop-valve can be closed quickly in case of necessity. A centrifugal spring governor is installed on the forward end of the rotor shaft. This operates a trip gear which disengages a rachet that holds the stop-valve open and allows the valve to close when the revolutions of the rotor exceed the maximum number of revolutions allowed. These turbines are tested for a speed of 20 per cent in excess of the rated speed.

Lugs on each side of the lower half of the casing are secured by holding-down bolts to the engine foundations, the holes for the forward holding-down bolts being oblong to allow for expansion.

TURBINE DATA.

Number of rotor wheels in ahead turbine	2
astern turbine	1

First Ahead Wheel.

Number of blades, first row	160
second row	213
Blade opening, first row, square inches	21/32
second row, square inches	1
Length of blades, first row, inches	1-35/64
second row, inches	1-23/32
Thickness of blades at base, first row, inches	.267
second row, inches	.204

Second Ahead Wheel.

Number of blades, first row	160
second row	213
Blade opening, first row, square inches	21/32
second row, square inches	1
Length of blades, first row, inches	1-35/64
second row, inches	1-23/32
Thickness of blades at base, first row, inches	.267
second row, inches	.204

Astern Wheel.

Number of blades, first row	100
second row	132
Blade opening, first row, square inches	21/32
second row, square inches	1
Length of blades, first row, inches	1-35/64
second row, inches	1-23/32
Thickness of blades at base, first row, inches	.267
second row, inches	.204

REDUCTION GEAR.

The reduction gear is of the planetary type, similar to those installed in the Eagle boats. This reduction gear is quite different from our usual conception of a marine reduction gear, in that the driving-pinion is inside of the gear instead of on the sides, and all shafting is in the same straight line. The driving-pinion drives two planetary gears (or rollers). These mesh with a stationary gear in the casing. These planetary gears are secured in a frame, and their rolling causes the frame to revolve. The frame is secured to the low speed shaft by a collar and key, and drives the propeller through the low-speed shaft.

A photograph of the reduction gear, with planetary cage removed, is here shown (marked Figure 2), which shows very plainly the principle of the gear.

The reduction gear casing is of aluminum, to reduce the weight, and the lower half is part of the turbine casing.

The reduction gear casing contains the reduction gears, gear cage, sun-pinion shaft and bearing (which bearing is also the after turbine bearing), planetary cage bearing and a Kingsbury thrust block. There are also contained in this casing forced lubricating oil pump, strainer and oil cooler.

An internal right and left hand helical gear of 11.8125 inches pitch diameter with 189 teeth is set in a recess in the gear casing and is keyed in place.

Revolving inside, and in mesh with the internal gear, are two planetary gears with same type teeth. The planetary gears are 5¼-inch pitch diameter and have 84 teeth revolving in bearings in the planetary gear cage, and mesh with the sun-pinion gear.

The planetary gear cage is connected to the low-speed shaft by extending over its forward end, and is keyed to it and drives the propeller.

A Kingsbury thrust block is fitted on the low-speed shaft in the after end of gear casing to take the thrust of the propeller.

All bearings in turbine and gear casing are of the self-aligning, spherical seat type.

Contained in the gear case is a small reciprocating pump driven by an eccentric on the low-speed shaft, and oil-cooler with necessary strainers and valves fitted. There is also an independent forced lubricating pump installed, to be used in case of break-down of this attached pump. A pressure of about ten pounds is carried on the gear cage bearing. The oil lubricates the bearing, then passes through a hole in the shaft and goes forward and passes out through holes in the sun-pinion and lubricates the planetary gears. Pipes lead from the pump to the other bearings.

REDUCTION GEAR DATA.

Material.

Gear casing	Aluminum.
Planetary gear cage	Cast steel.
High speed shaft	Chrome Vanadium steel.
Low speed shaft	Forged steel.
Planetary gear	Special steel.
Internal gear rings	Special steel.
Planetary gear pins	Forged steel.
Oil glands	Bronze.

Gear Data.

Internal gear rings, number of teeth	189
Planetary gears, number of teeth	84
Sun pinion gear, number of teeth	21
Internal gear, pitch diameter, inches	11.8125
width, each, inches	2.125
Planetary gear, pitch diameter, inches	5¼
width, each, inches	2
Sun pinion gear, pitch diameter, inches	1.3125
width, each, inches	2
Helix angle of teeth, degrees	23
Ratio of reduction	10:1

Rates of Turning.

Turbine shaft, R.P.M.	8,000
Sun pinion shaft, R.P.M.	8,000
Planetary gear on pins, R.P.M.	2,000
Propeller shaft, R.P.M.	800

Shaft Data.

Material of tail shaft	Rolled Brass.
Tail shaft, length, feet and inches	13–9
Rake per foot of shafting, inches	1½
Diameter of tail shaft, inches	2½

The tail shaft is supported by the stern tube bearing and strut bearing.

Propeller Data.

Material	Manganese Bronze.
Number of blades	3
Pitch, feet and inches	1–10⅞
Diameter, feet and inches	2–4
Projected area, 3 blades, square feet	2.649
P.A. ÷ D.A.	.619

MAIN CONDENSER.

There is one main condenser located in the engine-room on the center line between the turbines. It is a surface condenser built at the Navy Yard, Norfolk, and is designed to maintain a vacuum of 25 inches at full speed.

Data.

Cooling surface, square feet	231
Distance between tube sheets, feet and inches	3–6¼
Thickness of tube sheets, inches	9/16
Length of tubes as installed, feet and inches	3–7⅞
Number of tubes	403
Diameter of tubes, outside, inches	5/8
Thickness of tubes	No. 12 B.W.G.
Length of condenser, including heads, feet and inches	5–2¼
Inside diameter, feet and inches	1–9

The vacuum is obtained, and the condenser drained, by a 4½ inches × 6 inches × 8 inches vertical simplex, feather-weight air-pump, situated on the forward engine-room bulkhead. The condenser is fitted with a 7-inch nozzle leading from the exhaust side of each turbine, and has a 2-inch drain nozzle to the main pump.

The condenser is single flow and the tubes are expanded on each end. The circulating water is supplied by 4½ inches × 6 inches × 6 inches vertical simplex pump, when standing by or backing, through a 3-inch nozzle fitted on the forward end of condenser. When going ahead, the water is supplied by a scoop fitted under the boat through an 8-inch nozzle fitted to the forward condenser head. There is a 6-inch gate valve in the line from the scoop to the condenser, which is used only when the launch is secured for some time. There is a swing check valve installed aft of the gate valve referred to above, which is used when the launch is maneuvering and it is necessary to shift from the scoop injection to the circulating pump.

FEED TANKS.

There is a feed tank made of sheet copper, tinned on the inside, on each side of the launch extending from frame 29 to frame 33, holding 60 gallons each. There is also a feed tank of copper, tinned inside, holding 66 gallons, installed in the bilges under the boiler. This is installed in this place in order to prevent the high temperature of the furnace bottoms from coming in contact with the wood of the hull.

The feed tank on the starboard side is fitted with loofa sponges and acts as a filter tank. The tank on the port side is used only as a reserve feed tank.

FEED SYSTEM.

There are two Worthington vertical simplex pumps, 4 inches × 2½ inches × 4 inches, installed on the port side of the engine-room for feed pumps. The pumps take their suction through a manifold from the tank under the boiler and discharge through a coil feed water heater containing 5.4 square feet of heating surface, through a check and stop valve on the boiler.

There is a by-pass fitted around the feed water heater.

FORCED LUBRICATING SYSTEM.

There is situated in the gear case a plunger pump attached to an eccentric on the low-speed shaft, 1½ inches × ½ inch, which takes its suction through a strainer from the drain tank located in the base of the reduction gear case. In the after end of the gear case is an oil cooler of approximately 3 square feet of cooling surface of ⅜-inch U tubes. Cooling water is supplied from the circulating pump. Each system holds about ten gallons of oil.

BOILERS.

There is one type P oil-burning boiler situated in an air-tight compartment, just forward of the engine-room. The boiler is of the general design of the Bureau of Steam Engineering small-tube boiler, and consists of one steam drum and two mud drums connected by 1-inch tubes. There are six rows of from 25 to 27 tubes each on each side of the boiler.

This type of boiler was built at the Navy Yard, Philadelphia, and is equipped with one Modified Fisher Continuous-flow Burner which was developed at the Fuel-Oil Testing Plant especially for this boiler. The burner and all valves

connecting with it are operated either from the engine-room or from the deck, so that it is not necessary for a man to be in the fire-room when steam is on the boiler.

BOILER DATA.

Number of boilers	1
Working pressure, pounds, square inch gauge	250
Height to top of drum, feet and inches	5–10
Width to outer edge of lower drums, feet and inches	4–8⅛
Length to end of drums, feet and inches	6–2
Number of furnaces	1
Furnace volume, cubic feet	30.35
Heating surface, square feet	500
Capacity when full, gallons	184
Capacity at steaming level, gallons	144
Kind of draft	Closed fire-room.
Number of burners	1
Diameter of steam drum, inches	20
lower drums, inches	12½
main steam stop valves, inches	2¼
auxiliary steam-stop valves, inches	1
feed stop and check valve, inches	1
Height of funnel above bottom of furnace, feet and inches	9–3
Air pressure carried for full speed, inches water	1.2
Type of burner	Continuous flow.
Number of tubes	304
Diameter of tubes, inches	1

OIL-BURNER.

The burner is of the modified Fisher continuous flow type whereby a continuous flow of fuel-oil is obtained by the tip of the burner and any excess of oil above that required for the furnace is returned to the suction of the fuel-oil pump. The pressure is controlled by the opening and closing of the valve on the return to the suction of the fuel-oil pump.

Arrangement is made for steam atomization when raising steam so that a steam hose can be led from the ship to the launch to avoid smoking when alongside the ship.

BLOWER.

There is one turbine-driven Sturtevant blower on the bulkhead between the engine-room and fire-room. The turbine is on the engine-room side of the bulkhead and the blower is on the fire-room side. The blower takes its suction from a ventilator on the engine-room, and discharges into the fire-room against a pressure of 2 inches of water.

BLOWER DATA.

Maker	Sturtevant.
Type of turbine	Sturtevant T.V.-6.
Type of blower	Sturtevant No. 2, M. V. Des. No. 2.
Speed, R.P.M.	1,700
Pressure of air, inches of water	2.0
Number of nozzles	1
Number of bearings	2
Diameter of steam inlet, inches	½
Diameter of steam exhaust, inches	1¼
Capacity of blower, cubic feet per minute	2,000

MAIN STEAM PIPING.

The main steam line runs direct from the main steam-stop valve on the boiler to the throttle valve ahead of the maneuvering valves and supplies steam to the turbine only.

AUXILIARY STEAM PIPING.

The auxiliary steam line comes from the auxiliary steam-stop valve on the boiler, and after it enters the engine-room it divides into two branches and supplies the auxiliaries on the port and starboard sides.

FUEL-OIL SYSTEM.

There are two Worthington vertical simplex fuel-oil pumps 2½ inches × 1½ inches × 3 inches arranged to take suction from either starboard or port tank and discharge to the burner. There are two fuel-oil tanks, one on each side of the boiler, holding 150 gallons each, giving a total fuel-oil capacity of 300 gallons.

TEST OF ENGINE BEFORE INSTALLATION.

In January, 1919, a test was run on the first of these units at the works of the Poole Engineering Company at Baltimore, Maryland, in the presence of a representative of the Bureau of Steam Engineering. Trouble was experienced in obtaining the proper steam pressure as sufficient boiler power was not available. A small steam launch boiler was loaned by the Bureau of Steam Engineering for the purpose, but even with this boiler, dry steam of 250 pounds pressure could not be obtained. It was necessary to reduce the results obtained from the test to the standard guaranteed condition of the turbine.

STANDARD CONDITIONS.

S.H.P., each unit	100
R.P.M., turbine	8,000
R.P.M., propeller shaft	800
Steam condition, pounds per square inch gauge, dry and saturated.	250
Vacuum	25-inch mercury.
λ	173,000

TEST DATA.

Test No.	Steam Pressure Gauge per sq. inch	Quality of Steam	Condensate pounds	Charged to Auxiliaries	R. P. M. Propeller Shaft	SHP	Actual condition lbs. water per hour per S. H. P. all machinery	Reduced to standard condition lbs. water per hr. per SHP of turbine
1	192.4	0.88	1,700	255	797.4	82	20.1	14.8
2	187	0.855	1,700	255	800	81	21.0	14.8
3	202.8	0 907	1,730	260	800	82	21.1	15.4
4	183.2	.....	1,330	200	797.5	79	16.9	.....
5	178.7	0.912	1,532	230	800	72	21.3	15.4
6	160	0.887	1,432	215	798	64	22.4	15.9

TRIALS.

The trials were conducted off Norfolk, Virginia, on April 19-20-21, 1920, and consisted of the following:

Standardization trials;
Full power and endurance runs;
Test of auxiliary machinery, for water consumption.

The standardization trials were conducted over the measured mile in the main ship channel between Norfolk, Virginia, and Fortress Monroe, Virginia. Fourteen runs in all were made, as follows:

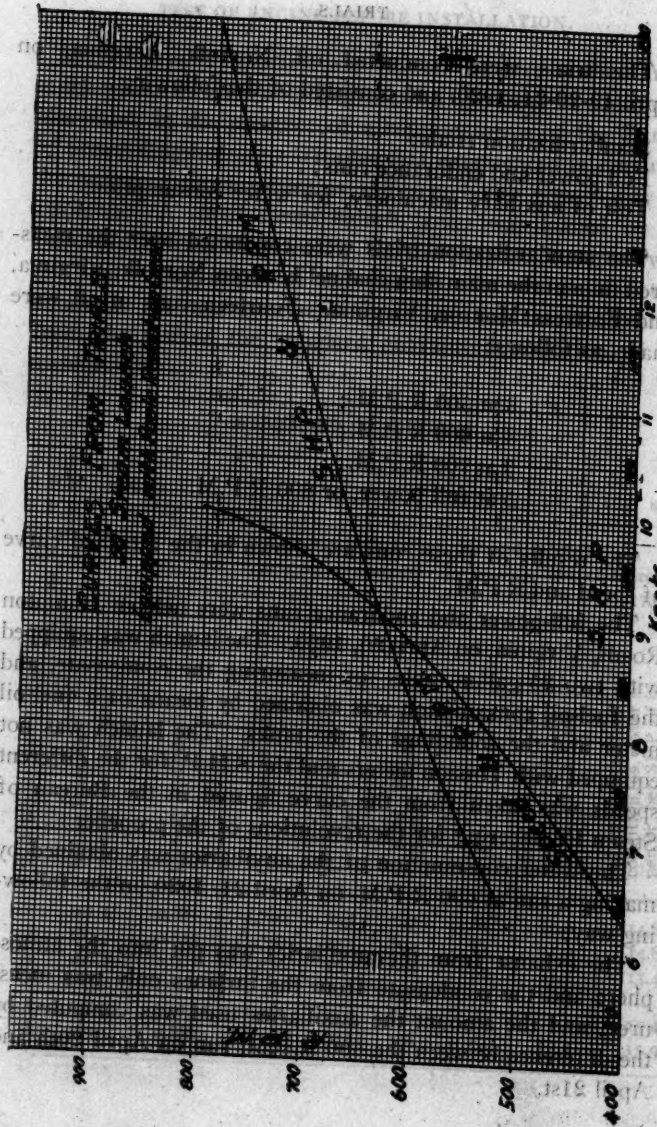
3 at 500 R.P.M.;
3 at 600 R.P.M.;
3 at 700 R.P.M.;
5 at full power, or 800 R.P.M.

The results of these runs are shown in the attached curve of speed and R.P.M.

The full-power and endurance runs were held in Hampton Roads, Virginia, on April 20, 1920. The launch was equipped with two 25-gallon tanks for measuring the condensate, and the fuel-oil consumption was obtained by means of a fuel-oil meter and the soundings of the tanks. The launch was not equipped with Torsion meters and the S.H.P. for the different speeds was taken from the curve figured in the Bureau of Steam Engineering for the revolutions of the propeller.

The water consumption of the auxiliaries was obtained by making a run at 600 R.P.M. on April 21, 1920, in the following way:

The exhaust from the auxiliaries was put into the atmosphere and the condensate from the turbines only was measured, and the amount the auxiliaries used was computed by the difference between the 600 R.P.M. run on April 20th and April 21st.


Knots
900
800
700
600
500
400

TRIAL DATA.

Kind of trial	Endurance run.	Full power.
Date	April 20.	April 20.
Length of trial	2 hours.	1½ hours.
Speed in knots, from curve	8.81	10.00
R.P.M., average	606.5	794.6
Pressure (average), main steam, pounds per square inch gauge	244	250
Pressure (average), turbine, pounds per square inch gauge	120	232
Vacuum, inches	25.3	25.7
Barometer, inches	29.80	29.80
Main feed, temperature, degrees F	188.5	170
Fuel-oil pressure, discharge to burner	189.4	225
return	106.7	174
Auxiliary exhaust, pounds per square inch gauge	5.5	4.0
Forced-draft blower, R.P.M.	1,533	1,609
Draft in fire-room, inches of water	1.0	1.28
Water consumption, total, pounds	6,175.8	7,020
Auxiliary steam, pounds	802	1,000
S.H.P., from curve	75	199
Pounds of water per S.H.P. per hour	30.4	18.49
Fuel, total pounds	540.8	569.7
Pounds of fuel per S.H.P. per hour	3.60	1,908
Pounds of fuel-oil, per knot run	30.74	37.94
Kind of fuel-oil used	Navy Standard.	Navy Standard.

A PECULIAR CASE OF DAMAGED PROPELLER.

By Lieutenant-Commander S. F. Heim, U. S. N.

The accompanying plates show the starboard propeller of the U. S. S. *Charleston* when that vessel went into dry-dock after steaming about 30,000 miles with this propeller in the damaged condition shown. The peculiar feature of the case is that the damaged propeller was just as efficient, if not more so, than the original propeller. It shows that in case of damage of this kind it is not always necessary to immediately renew or repair the propeller.

The facts of this case were as follows: The propeller was damaged by a pilot's tug fouling it when the vessel shoved off from Colon at the beginning of the World War; the tug was ripped open and had to be grounded to prevent its sinking; and each blade of the propeller had approximately eleven inches of its tip broken off and the ends of the blades were badly twisted.

The propelling efficiency of the ship was apparently unaffected by the accident as the "turns per knot" were the same after the accident as before. This was proved by many calculations from the navigator's data on long runs and also by actual standardization on the measured mile off Guantanamo at speeds of nine, twelve and fifteen knots.

The starboard engine required slightly more I.H.P. to make its revolutions than did the port engine, the propeller of which was undamaged.

The only detrimental effect observed was a slightly excessive vibration of the stern of the vessel when turning with full helm.

The blades were repaired at the Navy Yard, Boston, Mass., by cutting off the old tips, casting new tips and welding them to the old blades by the oxy-acetylene process.

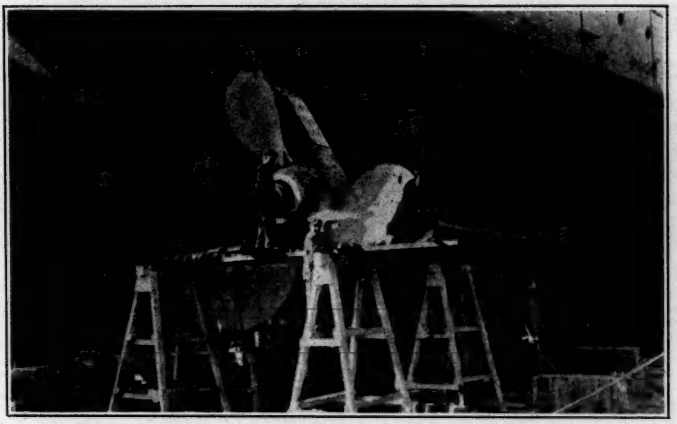


PLATE A.

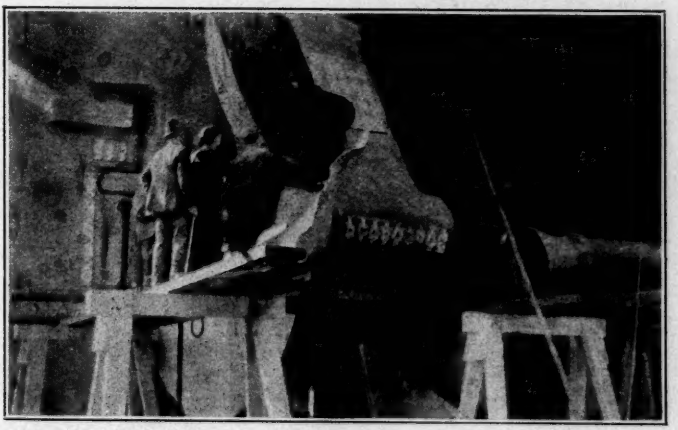


PLATE B.

The writer, as engineer officer of the vessel, was greatly interested in the case, because "knots per ton of coal" was so important on this type of vessel making convoy trips across the Atlantic and back, without coaling. It was thought that the repaired propeller would show increased efficiency, but, in spite of increased efficiency of personnel and plant, the desired increase of "knots per ton" failed to materialize after the blades were repaired. In fact, the accident seemed to increase rather then decrease the efficiency.

The reason for this erratic action can be explained by the following theory: The accident bent the blades so as to increase the pitch. The tips of the original blades turned too close to the hull of the vessel where they encountered the heavy hull wake current and thereby lowered the efficiency. The loss of area of the damaged blades was made up by increased pitch and the friction of their rough edges was about equal to the friction of the lost tips in the wake current near the hull.

NOTES.

THE STANDARDIZATION OF HORSEPOWER MEASUREMENT.

In compiling the returns of the horsepower of machinery built by the different marine engineering firms during the past year, a difficulty was revealed which did not exist before the war, when last the returns were made up at the end of the year 1913. At that time, nearly the whole of the propelling machinery for merchant ships was of the reciprocating type and the output of any firm was measured by the indicated horsepower of the machinery built during the year. The figures for the year just completed have shown that the geared turbine has largely replaced the reciprocating engine, although many firms are still engaged in building machinery of the latter type. But in presenting the returns, while the amount of reciprocating machinery is expressed as before in terms of indicated horsepower, the amount of turbine machinery turned out cannot be estimated in terms of indicated horsepower, since, so far, no simple method has been devised to measure the indicated horsepower of the steam turbine. Again, there is a growing amount of mercantile tonnage which is being propelled by internal-combustion engines, and here the estimated power is usually expressed, not as indicated horsepower, or as shaft horsepower, but as brake horsepower, which, while not differing greatly from shaft horsepower, differs very considerably from the indicated horsepower. In addition there is the possibility of the electric drive becoming more widely adopted for merchant tonnage, and in this case the power of the machinery is expressed, not in terms of the power of the turbines at all, but usually in terms of the electrical horsepower supplied to the motors. It is apparent, therefore, that with four classes of machinery being adopted for marine purposes and the power of each of these being computed in a different way, it is becoming more and more necessary, if accurate comparisons are to be made of the relative amounts of machinery of the different types built or of the size of the machinery of the different types required to give the same result on a ship of given dimensions, that some standardized method of measuring the horsepower of the machinery should be agreed upon, no matter what the type of propelling machinery adopted may be.

The indicated horsepower is certainly the simplest form in which the power of the reciprocating engine can be expressed. If the dimensions of the cylinders are known, with the links set so that the steam is cut off at a given fraction of the stroke, knowing the working pressure in the boilers and assuming a given number of revolutions per minute of the engine, the indicated horsepower can be calculated. Again, this figure can be readily checked by the simple expedient of taking indicator diagrams when the engine is actually in operation, and the diagrams thus obtained also serve the useful purpose of showing whether the valve gear is correctly set. Hence, the power of the reciprocating unit is always expressed in terms of indicated horsepower, although actual indicating of the engine may never be carried out.

With the steam turbine, however, there is no direct way of computing the power developed. By taking into account the size of the steam passages and knowing the working pressure, some idea of the power likely to be developed can be obtained, but this can only be verified when the turbine has been installed on the ship by taking a measurement of the shaft horsepower by means of a torsion meter. With this instrument, which has really been designed as a means of measuring the power developed by the marine steam turbine, the angle of torsion of a given length of the propeller shaft is accurately determined, and from this, the horsepower being transmitted through the shaft can be computed. But the apparatus required is by no means simple, and, while it may be installed temporarily during the trial trips of a vessel, it is not usual for it to be permanently in place, and therefore no direct method is available for determining, at any time when the vessel is at sea, the power being developed by the engines. It must be remembered, however, that when once measurements have been made with the torsion meter of the shaft horsepower developed by any particular type of turbine, the power to be expected from similar turbines, even if they are of different size, can be calculated with a considerable degree of accuracy. In any case the shaft horsepower is the simplest and most accurate form in which the power of the turbine can be expressed.

With the oil engine the most usual method of expressing the power developed is in terms of brake horsepower. This is, no doubt, because the engine is even now generally built in small sizes and the testing of the engine in the shop by means of a brake gives the simplest and most direct method available for determining the power developed. Quite large engines are, therefore, tested in this way, and when once the power of any type of engine has thus been determined, the power of similar engines of different size can be computed. It is possible, of course, to take indicator diagrams from the Diesel and the semi-Diesel engine, although this is not quite so simple as with the steam-reciprocating engine, and hence the indicated horsepower of the oil engine could, if found desirable, be obtained. With electric motors, the determination of the power supplied is a matter of great simplicity, it being necessary merely to instal an ammeter and a voltmeter in the circuit. This can be made permanently recording, and allowing for losses in the motor the power available can be always calculated. In order to get the exact output of the motors, however, they would either have to be tested by means of a brake, or for large powers, a torsion meter would have to be employed.

It is apparent, therefore, that if these various methods of expressing the power of the prime mover are to be replaced by one simple method, the shaft horsepower is the figure which should be agreed upon. With the reciprocating engine, the indicator would still be employed to show the behavior of the engine, but torsion meter tests could also be carried out, thus yielding a means of determining the mechanical efficiency of the engine and thrust-block, as well as giving a basis of comparison with engines of other types. With the turbine, the torsion meter method of determining horsepower is the only one available, and the shaft horsepower represents the power available for driving the propeller after losses in the gear and the thrust-block have been deducted. With the oil engine the brake horsepower is not greatly different from the shaft horsepower, the difference representing the losses at the thrust-block and one or two of the tunnel shaft bearings. Here, again, the indicator could still be retained and frequently utilized. With electric drive provided, the shaft between the motors and the stern tubes is not too short, torsion meter tests could be

readily carried out, the difference between the figures thus obtained and the power supplied to the motor giving the losses in the motor itself and the thrust-block. Thus it is suggested that, while the present methods of expressing the power of different types of marine engines and motors be retained, the shaft horsepower should be determined or computed in each case, and this figure used for purposes of comparison and compilation when expressing the power of machinery built or projected.—"Shipbuilding and Shipping Record."

DIESEL ENGINE CASTINGS.*

SOME PRECAUTIONS NECESSARY IN ORDER TO ENSURE GOOD CASTINGS.

Every one who has had to do directly with the manufacture of Diesel engines is convinced that the metallurgical side of the question is one of vital importance.

In cast-iron, when it is cooling down from the molten to the solid state, the planes of crystallization group themselves perpendicularly to the surface of the external contour, that is to say, at right angles to the outside surface and in the direction in which the heat of the fluid cast-iron has passed outward from the body in cooling and solidifying. Every abrupt variation in the external contour, every salient and every re-entering angle, no matter how small, upon the surface of a casting, is attended with an equally marked sudden alteration in the arrangement of the crystals of the metal. The natural remedy for this, therefore, is to avoid all sharp angles, ribs and sudden deviations of sectional outline, so that the metal while being poured may be allowed to flow in natural curved lines.

Every advantage should be taken of any process in manufacture that will facilitate closeness of grain with its consequent close packing and greater cohesion of the crystals. It is, therefore, advantageous to cast the cylinder liners and pistons on end, with the breach end at the bottom, thus taking advantage of the static pressure of the metal in the mould. Denseners also have been used for improving the compactness of the piston head, but the use of these requires special care or hardness will be overdone.

Hardness is a point which requires very careful watching in this class of work, as a cast-iron piston, if too hard, has a tendency to split right across when first put to work, particularly if it is subjected to full load soon after starting. A tough iron therefore is more desirable. The properties of cast-iron which are essential to successful working of Diesel engine pistons and liners are: High tensile and other physical properties; metal which will readily take a high polish under working conditions and thus give good resistance to wear, and the ability to resist growth and cracking when subjected to the high working temperatures which are a feature of this class of engine. These conditions are only met with in cast-iron possessing the highest intercrystalline cohesion, and this is also associated with metal which gives the highest tensile strength. It is a remarkable and fortunate circumstance that all these desirable qualities are associated with cast-iron in proportion to its tensile strength. This naturally narrows the choice, and materially assists in determining the proper metal to be used.

* Abstract of paper read on January 30 by Mr. F. J. Cook before the North-East Coast Institution of Engineers and Shipbuilders.

The factors which tend to give a cast-iron having the highest physical properties are:—

(1) Chemical composition.
(2) Casting temperature.
(3) Rate of cooling.
(4) Microstructure.

It is essential to take into account the rate of cooling when considering the chemical composition of metal to be employed in any given casting.

CARBON.

The most important element in cast-iron is carbon. It is not too much to say that all the physical properties of the metal depend upon the quantity of this element present and its condition, the value of the other elements being in accordance with the effect they have on the carbon and the compounds it forms.

The author's experience leads him to the conclusion that maximum tensile strength is obtained with 0.6 to 0.8 per cent of carbon, and maximum transverse strength with 0.4 to 0.6 per cent, and with total carbon not exceeding 3.25 per cent.

SILICON.

The element silicon is always present, in proportions most variable, whilst the influence it exerts on the condition of the carbon present, and consequently on the hardness and fluidity of the metal, is most marked. As the quantity and condition of the carbon have been stated to be so important, whilst the effect of silicon on this element is so marked, it is clear that if a formula can be adduced that will show when these two elements are in the best proportions for any given class of work, a very important stage will have been reached. With this object in view, the author has adopted the following more or less empirical formula, based upon the amount of carbon in the pearlite, and has used it with marked success in practical working:—

$$Sc = \frac{C}{4{\cdot}26} - \frac{Sil}{3{\cdot}9}$$

where Sc = ratio of silicon to carbon,
C = *total* carbon in percentage,
sil = silicon in percentage.

The elements carbon and silicon are in the best proportion for Diesel-engine pistons and liners and all parts requiring highest tensile strength when the value of Sc = 0.76 to 0.82, and for water-cooled cylinder heads and castings requiring maximum transverse strength when the value of Sc = 0.83.

SULPHUR.

By many, sulphur is looked upon as the arch-enemy of the ironfounder; but although its presence makes the molten metal thick and sluggish, thus giving rise to blow-holes, this element, in fair proportions—say, 0.12 per cent—adds considerably to the wearing properties and strength of the metal. In every case, however, sulphury iron *must* be cast as hot as possible if trouble is to be avoided.

PHOSPHORUS.

When the proportions of total carbon and silicon are those required for maximum tensile strength—as in the formula given—the proportion of phosphorus may be as high as 1 per cent without seriously jeopardizing the tensile strength; in fact, with steam cylinders and castings requiring maximum strength and of complicated design the reduction of shrinkage and the additional fluidity of the metal due to this percentage will generally result in a sounder casting, with fewer initial contraction strains. Where the ratio of silicon and carbon is different from that which has been suggested, the quantity of phosphorus becomes one of the vital elements and must then be kept as low as possible.

The following table, which contains part of the information gleaned by the author from several years' study of British and Continental results, goes to show that where the silicon and carbon are in the desired ratios, the quantity of phosphorus present does not become so extremely important:

	Suggested by Hurst. Per cent.	German. Per cent.	Another German. Per cent.	Continental. Per cent.	Replaced by Per cent.	Scotch. Per cent.	Suggested by the Author. Per cent.
Phosphorus..	0.52	0.33	0.59	0.54	1.34	0.54	Not over 1
Manganese...	1.44	0.72	0.99	1.08	0.28	0.97	Not over .5
Silicon.....	0.9	0.83	0.91	0.92	0.8	0.77	0.76–0.82
Results.....	....	Good.	Bad.	Bad.	Good	Good.	*

In each of the instances cited—and many more could be given if necessary—the results in actual working bore no relation to either the phosphorus or the manganese content of the metal, but were closely related to the results suggested by the silico-carbon ratio.

MANGANESE.

Manganese has a tendency to harden the iron both directly and by causing the carbon to remain in the combined form. In the presence of much sulphur, however, it may act as a softener by uniting with that element and forming manganese sulphide, which rises to the top of the metal and passes away with the slag. The general properties of the metal—other than a tendency to chill—are not materially affected so long as the manganese is not in greater proportion than 0.7 per cent. For castings in the making of which denseners are employed on faces which have to be machined, it is not advisable to have more than 0.4 per cent; otherwise there will be a tendency to chilling. As manganese tends to the removal of sulphur and the formation of brittle, spikey crystals, which break off under rubbing, much of this element is not good for castings which are required to have good wearing properties when working under heated conditions.

To sum up, the proportions of the various chemical elements which the author has found to give good results for Diesel engine cylinder liners and pistons are:—

Total carbon	=	3 to 3.2 per cent.
Silicon	=	1 to 1.2 per cent.
Phosphorus	=	not over 1 per cent.
Sulphur	=	0.12 per cent.
Manganese	=	not over 0.5 per cent.

* Has never given star cracks, low tensile results, nor bad wearing qualities.

For cylinder covers, which require to be not quite so hard for wearing purposes, and in which metal with rather less contraction is desired, the silicon can with advantage be increased to 1.5 per cent and the manganese to 1 per cent.

After 20 years' experience with this class of metal, the author finds that the most advantageous amount of steel to be added to the mixture is 15 per cent. Experiments with varying amounts up to 40 per cent have been made, but with above 15 per cent the metal has a tendency to be too hard, which is very undesirable for this class of work. Further improvement will probably be gained by the addition of some of the rarer elements and in this direction vanadium-chrome or nickel-chrome seem to hold out some prospect of success.

SURFACE WEAR.

There is much room for investigation into the subject of surface disintegration and wear of cylinders and all rubbing surfaces of cast-iron. When cast-iron is machined by the ordinary cutting methods there is more or less tearing away of the crystals. The tearing out of the softer material produces cracks or indentations and leaves the harder constituents standing up in relief. The degree or fineness of the finish will depend upon the structure of the material and the speed at which the material was cut; generally speaking, the quality of the finish will be improved as the cutting speed decreases. The resistance to wear under working conditions depends upon how far the crystals will resist breaking off—thus forming a hard, gritty, abrasive substance which increases the wear—or will spread over and form a highly polished surface. Grinding of cylinder liners and piston rings, being less drastic and exposing less of the harder crystals, appears to be a desirable method of finishing these articles, provided every trace of emery be removed. Where this is not practicable, very slow cutting speed and fine feeds for finishing should be resorted to with the object of decreasing the wear.

ANNEALING.

Considering the enormous quantities of heat which the pistons and cylinder heads have to deal with, it is imperative that everything in the nature of cooling strains in the castings should be eliminated, so that the crystallization shall be in its natural formation, allowing free and ample expansion. As all castings are more or less subject to initial strains, it is desirable that those having to work under heated conditions should undergo some heat treatment or annealing. It is to be feared, however, that frequently more harm than good is done owing to a general lack of knowledge of what is desired and of the results generally accruing from heat treatment of cast-iron, and to the crude manner in which the heat treatment is performed.

Whilst heat treatment of cast-iron, even at a low temperature, affects the physical properties of the metal, a prolonged treatment at 780° C. is positively dangerous, as, among other things, the pearlite is then broken down and the generally desirable physical properties of the metal are destroyed.

Considering the cooling effects the cylinder liner receives when at work, and bearing in mind that it is less subject to initial casting strains, the author does not consider heat treatment of these parts necessary. It is essential that the heat treatment shall be carried out in a properly designed furnace.

Two useful workshop tests—with the class of iron suggested—for the guidance of the founders are: Impact test and casting temperature. Whilst impact testing is not usually applied to cast-iron, it is, however, beginning to be used, particularly on the Continent, in connection with Diesel-engine work.

DISCUSSION.

In the course of the discussion which followed the reading of the paper, the President, Mr. A. E. Doxford, said the contribution was of great interest to those who had had trouble with Diesel engine castings. He knew of one case, in pre-war days, where cast-iron cylinder troubles were giving great trouble and the maker took cast steel instead, whereas another maker with the same trouble changed from cast steel to cast-iron. They had, however, got past that stage and the many papers read recently showed they were leading up to more scientific methods in the manufacture of cast-iron.

Dr. J. H. Paterson considered that cylinder heads, pistons and linings were three entirely different things and should be treated differently. A great many failures, in his opinion, were due to the fact that designers of engines did not realize the difference in combustion which took place according to the fuel intended for the engine. As a point of interest he mentioned that many of the castings from captured German submarines came within Mr. Cook's formula. They were good castings, low in phosphorus and high in manganese.

Mr. M. Riddell said that one could not over emphasize the importance of method in the foundry apart from chemical composition. He showed a number of slides illustrating how iron of the same chemical composition showed entirely different structures. The main thing, he thought, was that founders should rely more upon their processes than upon chemistry.

Mr. Cook briefly replied and mentioned that he stood practically alone in his attitude on phosphorus and manganese, but he could say, after 20 years' experience, that with his ideas he could produce results which could not be excelled or even equalled.—"Shipbuilding and Shipping Record."

INTERNAL COMBUSTION ENGINES.

Examination of metals or alloys entering into the construction of German submarine engines.

Publisher's Note.—We publish below the results of chemical analyses and physical examinations of metals or alloys used in the construction of 2-cycle engines (built by the "Germania-Krupp Co.") for several German submarines or "U-Boats." These results were obtained from specimens furnished by "La Technique Moderne," with the authorization of the Minister of Marine, whom we wish to thank very heartily here; also Mr. A. Citroën and his laboratory chief, Mr. Godfroid. To come back to our subject, in any future study of this question of the materials of construction of German submarines, we will confine ourselves to collecting the results of research enterprises formed at our request, valuable to present-day projects.

The principal material analyses were derived from samples of the crown of the piston, cylinder head, cylinder liner, fuel needle, the scavenger valves and scavenger pumps, and the valves of the fuel pumps.

Following are the results that were obtained:—

1. Piston crown.
   (*a*) Chemical analysis.

Carbon	0.24	Phosphorus	0.065
Manganese	0.47	Nickel	Trace
Silicon	0.24	Chrome	.....
Sulphur	0.04	Tungsten	.....
Vanadium	Trace		

   (*b*) Physical test.
   Hardness, Brinell test, 140.
   The appearance under the microscope is that of homogeneous carbon steel that had received no heat treatment.
   (*c*) Conclusion.
   The metal is an ordinary steel of the quality so-called "half-mild," having an ultimate strength of 68,000 lbs. per square inch.

2. Cylinder head.
   (*a*) Chemical analysis.

Tin	10.0	Iron	Trace
Copper	87.9	Zinc	2.08
Lead	0.08	Phosphorus	0.041

   (*b*) Physical test.
   Hardness, Brinell test, 100.
   The micrographs show nothing in particular.
   (*c*) Conclusion.
   Phosphor bronze with good friction qualities of average hardness as obtained with raw materials of good quality.

3. Cylinder liner.
   (*a*) Chemical analysis.

Carbon	2.81	Silicon	1.88
Manganese	0.52	Sulphur	0.135
Phosphorus	0.52		

   (*b*) Physical test.
   Hardness, Brinell test, 164.
   The micrographic examination shows a gray cast-iron, in which the graphite is distributed very uniformly throughout the structure and in which the amount of combined carbon is about 1 per cent.
   (*c*) Conclusion.
   Gray cast-iron very high in phosphorus, a little high in carbon, of a mixture approximating cast-steel.

4. Fuel needle valve.
   (*a*) Chemical analysis.

Carbon	0.15	Phosphorus	0.044
Manganese	0.46	Chrome	Trace
Silicon	0.21	Nickel	.....
Sulphur	0.036	Tungsten	.....
Vanadium	Trace		

   (*b*) Physical test.
   Hardness, Brinell test, 163.
   The micrographic examination shows an extra-mild steel, not having been annealed after machining.
   (*c*) Conclusion.
   Extra-mild steel with an ultimate strength of 54,000 pounds per square inch, either rolled or drawn to a rough finish.

5. Scavenger valve seat.

(*a*) Chemical analysis.

Carbon	3.25	Silicon	1.60
Manganese	0.65	Sulphur	0.135
Phosphorus	0.40		

(*b*) Physical test.

Hardness, Brinell test, 105.

The micrographic examination shows a gray cast-iron in which the graphite is uniformly distributed throughout the structure and in which the amount of combined carbon is about 1 per cent.

(*c*) Conclusion.

Gray cast-iron of the same composition as the iron in the cylinder liner.

6. Scavenger valve.

(*a*) Chemical analysis.

Carbon	0.35	Phosphorus	0.038
Manganese	0.82	Chrome	Trace
Silicon	0.33	Nickel	
Sulphur	0.031	Tungsten	
Vanadium	Trace		

(*b*) Physical test.

The micrographic examination shows a metal with pearlite uniformly distributed with a few impurities.

(*c*) Conclusion.

Steel of excellent quality, of medium hardness, not heat treated.

7. Fuel spray valve nozzle plate.

The micrographic examination shows a steel of medium hardness, not heat treated.

8. Scavenger pump valve.

(*a*) Chemical analysis.

Carbon	0.72	Phosphorus	0.073
Manganese	1.23	Chrome	18.1
Silicon	0.48	Nickel	Trace
Sulphur	0.031	Tungsten	
Vanadium	Trace		

(*b*) Conclusion.

Steel with a high chrome content.

9. Valves of Fuel Pumps.

(*a*) Chemical analysis.

	Valve No. 1	Valve No. 2		Valve No. 1	Valve No. 2
Carbon	0.27	0.40	Nickel	5.42	5.56
Manganese	0.40	0.59	Chrome	19.5	17.5
Silicon	0.33	0.19	Tungsten	Trace	Trace

(*b*) Conclusion.

A non-corrosive metal known as "Nichrome" or "Chrome-nickel."

10. Anti-friction metal.

(*a*) Chemical analysis.

Copper	8.45	Antimony	20.8
Lead	5.5	Tin	65.2

11. Lubricating oil.

Density at 66 degrees F.	0.936
Viscosity "Engler" at 59 degrees F.	35.1
Viscosity "Engler" at 122 degrees F.	4.65
Fluidity "Barbey" at 59 degrees F.	.976 cu. in./hr.
Fluidity "Barbey" at 95 degrees F.	2.750 cu. in./hr.
Fluidity "Barbey" at 122 degrees F.	6.283 cu. in./hr.
Fluidity "Barbey" at 212 degrees F.	31.720 cu. in./hr.
Flash point, Pensky	365 degrees F.
Sulphuric heating	33.8 degrees F.
Freezing point	46.4 degrees F.

Trace of free carbon.
Reaction neutral.
Did not contain resinous oil.

12. Fuel oil.

Density at 66.2 degrees F.	0.861
Flash point	176 degrees F.
Viscosity, Engler, at 59 degrees F.	1.2
Viscosity, Engler, at 122 degrees F.	1.0

The above results prove that the "Germania-Krupp" concern has determined with particular care the composition of the materials used in the construction of Diesel engines for submarines, in spite of the difficulties of renewing their stocks in Germany. We will show in future issues that this care is one of the most essential conditions for ensuring reliable operation of this type of engines.—Translated from "La Technique Moderne," of November, 1919.

NON-FERROUS METALS AND ALLOYS AND THE NAVAL ENGINEER.*

ENGINEER VICE-ADMIRAL SIR GEORGE GOODWIN, K.C.B., LL.D., OUTLINES SOME OF THE DIFFICULTIES ENCOUNTERED IN NAVAL PRACTICE.

The inherent purpose of the Institute of Metals is to join together those interested in non-ferrous metals and alloys—that is, metallurgical scientists, manufacturers, and users of these materials—in research, in investigation of failure, and application for the advancement of the use of such materials. As one of the last-named class, it is appropriate that my remarks in this address should be mainly confined to the users' aspect of our work, and they will be given chiefly from the point of view of the naval engineer. That this is an important phase of the subject will be agreed, I think, when it is recalled that the British Navy is the largest individual user in the world of non-ferrous metals. The standards set by the Admiralty for most of the metals required by them have always been high, and to be on the Admiralty List for the supply of these materials has, I believe, always been regarded by manufacturers as a valuable asset.

ADMIRALTY STANDARDS.

It has sometimes been suggested that it is somewhat arrogant for the Admiralty to insist on such high standards, that it is, to say the least, extravagant to decree that the best and the best only is good enough for the Navy; but I can assure you that, although we do not always require

* Abstract of presidential address delivered at the Annual Meeting of the Institute of Metals, March 11.

the best, when we do ask for the best it is because the best is absolutely necessary, and, further, that if still better materials were available we should require them.

The position at present reached as a result of the cumulative endeavor during the last few decades is that in our latest battle cruisers we hope to obtain 144,000 s.h.p. on a total machinery weight (including water) of 4,750 tons, that is, at the rate of 74 pounds per s.h.p., and our most recent destroyers have over and over again developed over 28,000 s.h.p. on a weight of 32 pounds per s.h.p.

When the type of machinery is selected for a given ship, and the general features of the design are decided upon, the design in detail is proceeded with. This entails consideration of the dimensions, and reduction in weight is effected by increasing the stress allowed in the material as far as such stresses can be estimated or calculated. The Institute of Metals can take a large part, by itself and through its members, in assisting the development of naval engineering and other branches of engineering where light weight is the essential desideratum. It must, however, be realized that the problems are never ending; as soon as a particular object is reached new demands arise, and very often the job has to be tackled afresh.

NON-FERROUS METALS IN MARINE ENGINES.

The materials used in engine construction are chiefly iron and steel, on account of their high strength and relative cheapness. In marine engines, and particularly in naval machinery, non-ferrous materials are more extensively used on account of their greater resistance to the peculiar forms of corrosion associated with marine use. In certain cases such materials are used generally, on account of their better heat-conducting properties.

Our most important use of non-ferrous metals is in condenser tubes, and as we have these tubes at present in the Navy, we are, I think, fully satisfied with what our manufacturers, after years of patient endeavor, have given us, except that we want just a little more freedom from sea-water corrosion. The investigators working for the Corrosion Research Committee of the Institute are doing great work in this connection. They have already shown us how improvements may be expected, and the results of their labors are being turned to practical account. You will shortly have an opportunity of hearing their fifth report, which you will find, I am sure, of very great value. When they have succeeded in enabling us to remove from our minds doubts as to the freedom from corrosion of our condenser tubes, we engineers will assuredly be under an everlasting debt of gratitude to them.

Following their suggestion, a set of tubes, pre-oxidized under their system, was fitted in one condenser of a patrol vessel for comparison with a similarly manufactured set fitted in the other condenser without having been so treated. Unfortunately for the comparison, both sets of tubes have behaved themselves quite satisfactorily so far, so that the comparison is of little positive value. As far as our opinion is of interest, I may say that we believe in, and are strongly predisposed towards, the pre-oxidized tubes.

This trouble from corrosion of condenser tubes is the most annoying one experienced in the service; it is annoying because we feel we have got so near success, and also because of the serious consequences of even small leakages.

It has recently been recorded that the Grand Fleet was very much inconvenienced at times owing to leaky condenser tubes; but it must be re-

membered that this cuts both ways, and that ships of all navies suffered from this evil. There is sufficient evidence to justify the belief that proportionately others had to put up with more inconvenience than we experienced.

CONDENSER TUBES IN GERMAN WARSHIPS.

We have recently obtained specimens of tubes from some of the ships of the German Navy. A very noticeable feature of the samples examined is that most of the tubes are either tinned or lead lined. The tinning of tubes was a common Admiralty practice years ago when the circulating water was outside the tubes, and it was of material assistance in prolonging the life of the tubes. The external surface could be easily completely coated with tin, but such a complete covering is more difficult to obtain on the interior surface of a long tube of small diameter, even with the greatest care; apparently should a small spot be left uncovered an intense action is set up, greatly reducing the life of the tube. I suppose the same sort of thing would occur with lead-lined tubes and other troubles at the tube ends might reasonably be anticipated. For this reason such tubes have not had much use in our service, and have been totally abandoned for many years.

We, however, have great faith in a suitable covering of homogeneous quality, provided it is complete, such as the pre-oxidizing method promises to be, and provided it will stand the erosive effects of circulating water under conditions of normal service.

In the latest sample we obtained, from one of the newest light cruisers, there is no such lining or coating, and the tubes are practically of the British Admiralty composition with a very small quantity of iron.

The possibility of using tubes of composition different from those in general British practice has occupied the attention of many workers, and it is quite a legitimate field. It is well known to us that materials can be used which will be practically free from corrosion; but to all such workers a word of warning may be useful, and that is that freedom from corrosion is not the only requirement in a condenser tube. The tubes must not be too expensive. The manufacturers must be able to draw them practically and commercially, and they must comply with several other conditions which are fully laid out, I think, in the Admiralty specification, with which members are fairly well acquainted. In addition, I would also advise prospective workers in this field to read Dr. Hatfield's recent paper on season cracking, and reap the further benefit to be derived from a perusal of the ensuing discussion.

TURBINE BLADING.

The second example I will quote is turbine blading. For many years turbine blades were made of brass, which was convenient in manufacture. The peripheral speeds and consequently the stresses were comparatively low, and on the whole this material has given, and is continuing to give, good results in designs suitable for its use. With the introduction of impulse turbines difficulties became apparent. They may have been due to the increase in the steam temperature attending the employment of a small amount of superheat or to increased stresses on the blades of turbines of this type beyond the ordinary centrifugal stresses, or to a combination of the two. The higher stresses can be met to some extent by increased scantlings, but as is well known the ordinary brasses fall off rapidly in strength as they are exposed to higher temperatures.

Phosphor-bronze blading was therefore introduced for the high-pressure end of such turbines, and with the higher peripheral speeds of impulse turbines of the geared type, the use of blades in which the root and blade are combined has been much extended for these designs.

Tests of phosphor-bronze as used for this blading have shown an ultimate strength of 23 tons per square inch at ordinary temperature, and 20.7 tons at 440 degrees Fahr. This is satisfactory as far as it goes, but a material is sought which, while yielding nothing to phosphor-bronze in its other qualities, will give a higher tensile strength and safely permit of high peripheral speeds where this would be of advantage, or alternatively reduced blade scantlings with its attendant advantages in respect to the turbine weight.

Steel has been considered and, although a highly suitable material in most respects, it has not been considered acceptable owing to the known risks of rapid corrosion, under conditions almost peculiar to the naval service. The Ferranti method, in which a thin coating of nickel is drawn over the steel blade, is known to give blades which are in a satisfactory condition in a foreign cruiser after a number of years' use, and a few rows of such blades fitted experimentally in two torpedo-boat destroyers of our service have also proved satisfactory. This system appears, however, only suited to blades of uniform sections, and has not, so far as is known, been applied to the other types. Our remedy appears, therefore, to lie in a non-ferrous material. Monel-metal has many attractions, but we are not aware that the manufacturing and supply difficulties have yet been satisfactorily overcome, although it has been used on a small scale for the blading of turbine auxiliaries.

ALUMINUM-BRONZE FOR TURBINE BLADES.

Dr. Rosenhain has kindly brought to my notice information bearing on his work in connection with processes of blade manufacture, and with the suitability of aluminum-bronzes for blading purposes. Sample castings have been made to his suggested compositions and worked up by one of the firms specializing in turbine blade manufacture into what appears to be satisfactory blading; opportunity is awaited to test it under actual running conditions.

As is well known, there are many aspects to consider in connection with the suitability of material for blading work, apart from the matter of strength; but it may be of interest to the institute to learn that erosion, which appears to have been serious in some shore installations, has not yet troubled us.

With the increasing use of superheat, we shall require the assistance of metallurgists and manufacturers to give us materials capable of withstanding the high stresses, necessary with our limited scantlings, when exposed to high temperatures, and turbine blading will demand a great deal of attention. In addition it will be necessary to review the work already done in investigating the properties of gun-metal and other alloys when exposed to high stress and high temperatures, and in all probability will also be necessary to undertake several other new and similar investigations.

MATERIAL FOR PROPELLERS.

The propellers of our warships for some years have been universally made of manganese bronze, gun-metal propellers still existing in some of the old vessels. The practice of using expensive alloys has been criticized from time to time, but such special alloys are necessary for this purpose, as they possess the several qualities required better than other materials.

To obtain the best efficiency from a propeller of given dimensions, it is necessary that the blades should be thin and their surfaces polished. We should have some difficulty in obtaining thin blades in sound cast steel of the large dimensions required, and a polished surface in either cast or forged steel could not be maintained in sea-water. Forged steel blades have been fitted and tried in some of our smaller vessels, but serious practical difficulties attended their use and they were abandoned many years ago.

For some years a peculiarly severe action had been observed in the fast-running propeller blades of our light craft for which a cause could not be assigned. The defects which developed were at first attributed to foundry faults, perhaps with some justification, but when this factor was surely eliminated the effects continued. The action was termed "erosive," but the location of the defect and the nature of the deterioration of material left no doubt that the action was not erosion as the term is generally understood in marine engineering. In order to endeavor to find more truly the cause of this action a committee of investigation was formed by the Admiralty, which consisted almost entirely of members of this institute. Our late President, Professor Carpenter, and Sir Charles Parsons took a prominent part in the consultations and experimenal work, and with the confirmatory assistance of eminent mathematicians it was concluded that the action was due to enormous stress produced in local patches in the material set up by a sort of water-hammer action caused by the collapse of the boundaries of vacuous spaces in the water arising from cavitation. The stresses were such that no material in practical use could stand the shocks and the cure had to be found in quite another direction, *i. e.*, cavitation had to be prevented.

BEARING METALS AND LUBRICATION.

Very interesting papers were read and discussed at our last meeting at Sheffield on the subject of white metals—interesting because they showed that workers for different purposes were in the same field. To be quite candid, it is probable that the bearing metal problem has lost a great deal of the importance that formerly belonged to it, owing to the fact that the true principles of lubrication so long known but not utilized have now been applied in a manner which can properly be described as revolutionary. It has long been thought that the nature of the bearing metal itself was a minor matter, provided that the oil film between the rubbing surfaces could be maintained; and the problem before engineers for years has been how to maintain that film with the ever-increasing direct bearing pressures and rubbing velocities. Following the mathematical investigation of the problem by Professor Osborn Reynolds some years ago, the practical solution of the problem has been effected by Mr. Michell in no uncertain manner. It has been adopted unreservedly in the Navy for the thrust blocks of all descriptions of engines with complete success. Although some difficulties have presented themselves in the application of the principles to marine journal bearings on account of the need of reversibility, they seem to have been overcome, and a design is in successful operation in two of the turbines of one of our torpedo flotilla leaders, aggregating 40,000 s.h.p. on two shafts.

Some work of great importance, however, still remains for the metallurgist. We have had, not by any means many, but one or two failures, and the investigations that are in progress may indicate valuable improvements in our workshop methods of dealing with bearing metals. It is further to be borne in mind that, presuming it to be true that the material of

the bearing is not of primary importance while the lubricating film is maintained and actual contact of the rubbing metallic surfaces is avoided, immense mischief may be done in a very short time should the lubrication fail, and to provide for such contingencies it is still considered eminently prudent by engineers to provide in bearing materials for such mishaps, and to this extent the problem of the bearing material remains as important as it always has been.

NON-FERROUS METALS IN OIL ENGINES.

A final example of the use of non-ferrous alloys will be found in the oil engine.

In developing the fast-running heavy oil engines a limitation was set for some time to the piston speed, owing to the high inertia stresses consequent upon the weight of the reciprocating parts. This limitation has been to a considerable extent removed by the adoption of pistons of aluminum alloys, and it is probable that further developments in this direction can be effected when we are more fully aware of the properties of such alloys to withstand the effects of high stress and high temperature, and as they are developed in conjunction with engineering devices to give improved results in the matter of heat flow.

With the view of reducing weight we are further considering the use of lighter working barrels of the cylinders by the adoption of non-rusting steel, and this appears by experiments made in our laboratory to have been rendered practicable by the use of piston rings made of copper-zinc alloy, shaped in the form of obturator rings.

I have endeavored by means of a few typical examples to show the manner in which naval machinery has developed, and may be expected to develop, in the future as far as the interest of this institute is concerned, and the instances I have given show, I think, that all sections can cooperate to advantage. There has already been a great deal of coöperation, but I have a feeling that it has been to some extent in two pairs. I know that manufacturers have worked with engineers, and very successfully; I know that the metallurgist has worked with the manufacturer, and I believe with equal success. But the metallurgist has not been brought into direct touch with the engineer as much as is desirable, and if contact of this nature can be improved, it will be for the good of engineering science and progress.

It has been arranged, with the full concurrence of the professional chemists and metallurgists concerned, that our naval engineer officers now and in the future shall receive instruction in metallurgy, not for a moment intended to make them metallurgical experts, but to a sufficient extent to enable them to discuss their experiences and their difficulties on a much more equable basis than has hitherto obtained.—"Shipbuilding and Shipping Record."

THE CORROSION OF CONDENSER TUBES.*

The present report is confined to the corrosion of condenser tubes—mainly 70:30 brass—and is a study of the practical problems of corrosion in condensers under service conditions, employing either sea water or fresh water. It is divided into four sections.

* Fifth Report to the Corrosion Committee of the Institute of Metals. Official abstract.

The first section deals with what has been called the diagnosis of condenser tube corrosion. The procedure to be followed in withdrawing and preparing a tube for examination is described, also the symptoms or appearances within the tube which correspond to each of the five main types into which the practical problems of corrosion under fresh-water or sea-water conditions have been classified. The importance of additional information concerning (*a*) the water supply and (*b*) the corroded tubes towards elucidating the cause of corrosion is shown and emphasized. This information, which is specified, has rarely been obtainable in the past, particularly as regards the water supply, and this may partly account for the lack of appreciation of the importance in corrosion troubles of the conditions existing within the plant. These conditions frequently vary very much from time to time, and it is shown that although the conditions which favor accelerated corrosion may be present for but short periods at irregular intervals, and consequently may not be easily detected, the effect on the tubes may still be very serious. Also in certain cases it is shown that accelerated, localized corrosion may persist after the initiating conditions have disappeared.

The second section is devoted to a consideration of certain features in the structure of condenser tubes, since an appreciation of these is of great value in following the mechanism of the types of corrosion studied later. Attention is principally directed to the presence on the tubes of a surface layer consisting of structureless and highly distorted metal. This layer has undoubtedly a greater resistance to corrosion by saline and fresh waters than the underlying crystalline metal, so that whenever this layer is penetrated corrosion will proceed at an increased rate. The layer has been stripped from a number of tubes of different composition. Its thickness is usually of the order of 0.01 millimeters, and indications have been obtained that its composition may be somewhat different from that of the underlying metal.

In the third section the five main types of brass condenser tube corrosion are considered separately in detail.

Type I. General Thinning.—This type may be considered as an accelerated form of the complete corrosion which normally occurs in saline solutions, in so far as the tube is gradually and uniformly reduced in thickness. The rate of ordinary complete corrosion is too slow to be of any serious consequence in practice. Laboratory experiments with running sea water have shown that a period of fifteen to twenty years may elapse—even presuming that the rate of corrosion is uniform and does not decrease with time—before a tube is reduced 50 per cent in thickness from this cause. Tubes always fail in practice by local action of some kind long before ordinary complete corrosion has seriously reduced the thickness of the tube. Rapid general thinning, however, is essentially a fresh-water phenomenon, and is usually associated with the presence of free acid in the water supply. The results of tests on ten samples of tube of widely different composition in hydrochloric acid of concentration 3 parts in 100,000 at ordinary temperature show that in six weeks all tubes had lost from 2 to 4 per cent in thickness. That such a very dilute acid should reduce the thickness of a tube so much in such a short time at ordinary temperature is ample evidence of the serious effect of acid in the water supply upon the life of condenser tubes. Proper neutralization, preferably at the source of contamination, is an effective remedy for this trouble, but considerable difficulty may be experienced in detecting the acid, especially if, as is generally the case, it enters the water supply only intermittently. Regular and frequent tests of the water must be made whenever this type of trouble occurs.

Type II. Deposit Attack.—The principal cause of pitting which is the most frequent source of trouble in condensers is ascribed to what is termed "deposit attack." In the presence of sodium chloride solutions, the cuprous oxide formed on a brass surface gradually changes to cuprous chloride. The latter is usually swept out of a condenser tube by the circulating water, but under various conditions may adhere at different parts of the tube surface. When such adherence has occurred, conditions now allow of the further gradual change of the insoluble cuprous chloride under the influence of oxygen to soluble cupric chloride and cuprous oxide. The action of cupric chloride solution on brass is very rapid, as may be gathered from the fact that a piece of brass tube, 2 inches long, placed in a strong cupric chloride solution at ordinary temperature, was completely disintegrated and partially replaced by a pseudomorph in copper in two days. The action involves the oxidation of the copper and reduction of cupric chloride to cuprous chloride. Redeposition of copper from solution by the zinc also occurs. In the presence of air cuprous chloride will again be converted to cupric chloride and the attack on the brass continued. Thus the action is both recurrent and local. Foreign bodies, and particularly colloidal bodies, lying in the tube have an injurious effect by serving as loci for the adhesion of cuprous chloride and by preventing the diffusion of cupric chloride. Observations on the incidence and distribution of pits in condenser tubes are shown to agree with the results which would be expected to follow from the above explanation of the mechanism of pitting. Attention is drawn to the importance of keeping tubes clean and as free as possible from foreign bodies as a means of preventing deposit attack.

Type III. Layer Dezincification.—An account is given, preliminary to the consideration of this type of corrosion, of the mechanism of so-called dezincification. The conclusion is reached that true parting of zinc and copper in a 70:30 brass does not occur, but that the so-called residual copper is always redeposited copper. Dezincification is therefore only apparent and not real, and the term is always subsequently employed by the authors in this sense.

The layer type of dezincification, which is characterized by disintegration of the brass tube and redeposition of copper over large areas, has been found to occur under both marine and fresh-water conditions. Several ways in which such action may occur are indicated. Under fresh-water conditions, it is often associated with acid water, particularly if the acid is not too dilute and the access of oxygen is not very easy.

Type IV. Plug or Local Dezincification.—This type of local corrosion may be regarded as a form of deposit attack, as it always proceeds beneath a deposit, and is stimulated by the presence of foreign bodies. It differs, however, in many ways from Type II. So far as is known, it occurs only in sea water—or diluted sea water—and is always associated with adhering white salt, consisting of colloidal zinc oxychloride—also containing some carbonate. It is readily reproducible in the laboratory, as it occurs spontaneously on the surface of 70:30 brass tubes after immersion in sea water at elevated temperatures—40 degrees to 50 degrees C.—for a few days; and is hence rather more amenable to experimental study than the previous types of corrosion. It is shown that the production of the right concentration of zinc in the liquid layer adjacent to the corroding brass surface plays a large part in the formation of the characteristic white oxysalt, and that interference with the production of this condition, *e. g.*, by lowering the zinc content of the brass or by raising the zinc content of the sea water, is sufficient to prevent its occurrence. The dezincifying action is

thought to be due to a small concentration of hydrochloric acid contained within the colloidal white salt. The difference in behavior of different batches of 70:30 tube—some always showing local dezincification, others never—persists after annealing or pickling, or both, and the reason for the difference in behavior is still not clear.

Type V. Water-line Attack.—In the case of a brass tube only partially immersed in sea water, increased corrosion—compared to that of the immersed portion—takes place, not at the water line as is commonly supposed, but above it, and sometimes as much as 2 centimeters above the air-sea-water surface. Further, the attack is not uniform, but is concentrated at areas where salt deposits have formed and is coterminous with the area covered by the deposits. Narrow bands of salt connect the sea water with the deposits. This type of corrosion is obviously a special form of deposit attack, taking place under the most favorable conditions, inasmuch as the attack beneath the deposits is much more severe than in any of the previous types of corrosion. This type of attack may occur at the inlet end of condenser tubes when entangled air clings to the surface of the tube and is prevented by eddying effects from being swept away by the water flow.

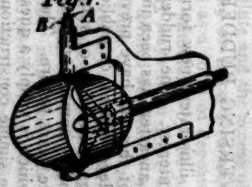
The fourth section of the report contains an account of preliminary work on the electrolytic protection of condenser tubes. The particular question investigated was that of the efficiency of electrolytic protection in preventing deposit attack, *i. e.*, attack by cupric chloride solution. A piece of 70:30 brass tube made cathode to strip iron in a normal cupric chloride solution is very distinctly attacked despite the small current passing from the solution to the brass. It is considered that an electric current slows down, but does not necessarily inhibit corrosion of a cathode. By raising the current density sufficiently it is possible that all corrosion, even in a cupric chloride solution, may be prevented. Experiments have shown, however, that a current as large as 20 to 25 amperes per 1,000 square feet is insufficient to prevent a 70:30 brass tube from being rapidly corroded by such a dilute solution of cupric chloride as a one-twenty-fifth normal solution. Some suggestions are made whereby the usefulness of the electrolytic protection process may be extended by special manipulation of it in the early part of the life of a tube, with the object of forming a thin continuous layer of calcium carbonate over the surface of the tube.—"The Engineer."

THE KITCHEN REVERSING RUDDER.

On this and the following pages we illustrate a new form of ship's rudder which is of very considerable interest. The rudder, which is the invention of Mr. J. G. A. Kitchen of Lancaster, has been in use for several years, but considerations arising out of the war have prevented, up to the present, either its proper commercial development, or publication of particulars relating to it. The rudder belongs to the class of simple and effective devices which, once they have been applied, immediately give rise to wonder why the application had not been made years before. The whole arrangement, apart from the operating gear, which is simply a question of mechanical design, is shown in Fig. 1, and the simple construction there shown is sufficient to enable a boat to be steered, reversed, controlled in speed and maneuvered in any way from a single tiller, with the engines running continuously in the forward direction. The advantages of being able to do this will vary in different cases, but without enlarging on the matter at the

SHEWING RUDDERS IN PERSPECTIVE.

Fig. 1.

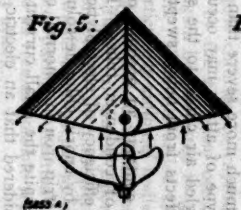


POSITION FOR FULL SPEED AHEAD. RUDDERS PARALLEL.

Fig. 2.

POSITION FOR FULL SPEED ASTERN. RUDDERS CLOSED.

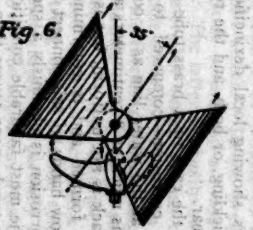
Fig. 5.



(BASS &c.)

POSITION FOR FULL SPEED AHEAD TO PORT (OR STARBOARD)

Fig. 6.



POSITION FOR HALF SPEED AHEAD. RUDDERS PARTLY CLOSED.

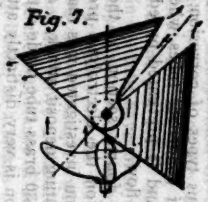
Fig. 3.

POSITION FOR REMAINING STATIONARY. RUDDERS PARTLY CLOSED.

Fig. 4.

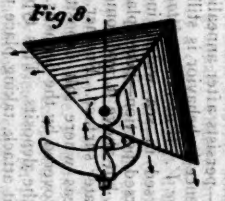
POSITION TO REVOLVE SHIP WITHOUT PROGRESSION

Fig. 7.



POSITION FOR GOING ASTERN TO PORT. (OR STARBOARD)

Fig. 8.



moment it is clear that complete and direct control from the bridge over all movements during maneuvering is obtained and further that all engine reversing gear may be eliminated. This latter point is of direct and immediate interest in connection with craft driven by internal-combustion engines.

Before dealing further with the application of the rudder, it will be well to say a little more in detail in reference to its construction. As shown in Fig. 1, it consists of two curved deflectors forming parts of a circular cylinder and partly enclosing the propeller. The deflectors are pivotted at the

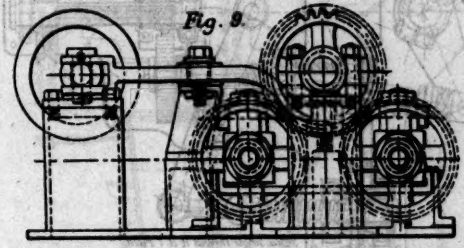


Fig. 9.

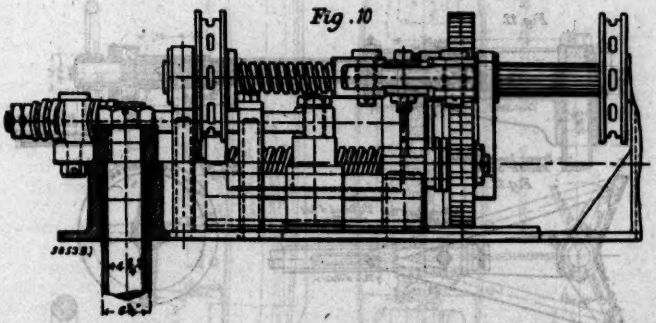


Fig. 10.

top and bottom on common centers, and are capable of being swung together in the same direction, or equally in opposite directions. Some of the possible positions of the two deflectors are shown in Figs. 2 to 8. One deflector is operated by the solid shaft A and the other by the hollow shaft B, through which A passes, as shown in Fig. 1. In ordinary running ahead the rudder would be as shown in Fig 2, the deflectors standing fully open and entraining the propeller race. Steering is carried out as with an ordinary rudder, the two deflectors moving together into the position, for example, shown in Fig. 6. Speed control is achieved by altering the angle between the deflectors. When they are partially closed, as shown in Fig. 3, part of the stream of water coming from the propeller escapes between the deflectors and still drives the boat forward, but part is deflected forwards and tends to have the opposite effect, with the result that the boat is slowed up, although the engine speed is unaltered. By further closing the

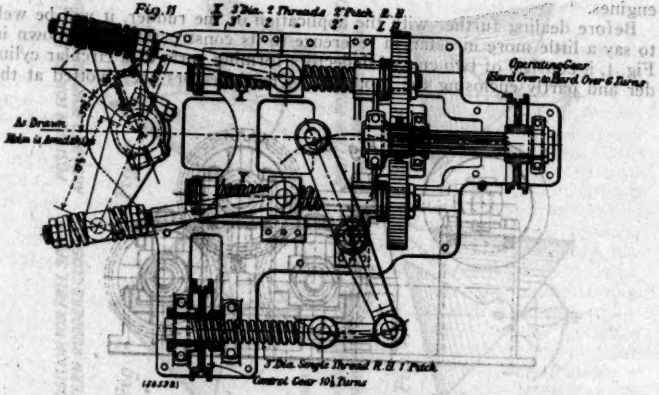


Fig. 11
Operating Gear
Hand Over & Hand. Over 6 Turns
Control. Gear 10½ Turns

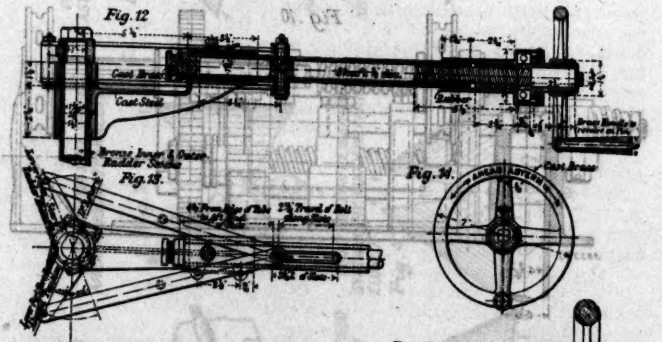


Fig. 12
Fig. 13.
Fig. 14.

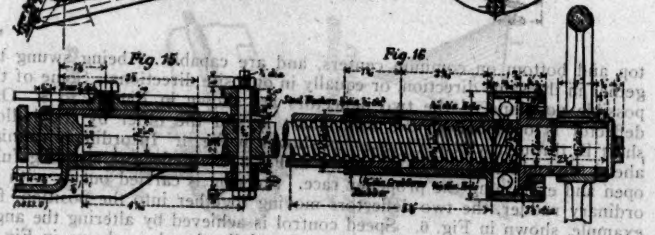


Fig. 15.
Fig. 16

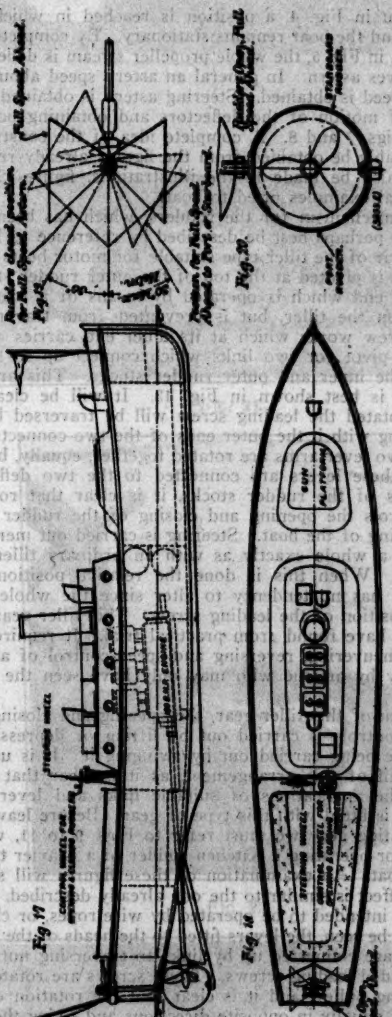


Fig. 17

Fig. 18

APPLICATION OF THE KITSEN REVERSING RUDDER.

rudder, as shown in Fig. 4, a position is reached in which the two reactions balance and the boat remains stationary. By completely closing the rudder, as shown in Fig. 5, the whole propeller stream is deflected forwards and the boat moves astern. In general an astern speed about one-third of the full ahead speed is obtained. Steering astern is obtained by combining the two types of motion of the deflectors and obtaining positions of the type shown in Figs. 7 and 8. A complete idea of the construction of the rudder will probably be obtained from the figures already referred to, but reference may also be made to the illustrations, figures 21, 22 and 23, which show actual examples fitted to boats.

The operating mechanism for the rudders, which has been very prettily worked out, may perhaps best be described by reference to Figs. 12 to 16. These show a gear of the tiller type suitable for motor boats and launches. The tiller, which is pivoted at the top of the outer rudder stock, carries a nut at the outer end which is operated by means of a handwheel. The nut can rotate on the tiller, but is prevented from traversing. In the nut a leading-screw works which at its inner end carries a vertical bolt which acts as a pivot for two links which connect up to levers secured respectively to the inner and outer rudder stocks. This arrangement of links and levers is best shown in Fig. 13. It will be clear that if the hand-wheel is rotated the leading screw will be traversed backwards, or forwards, carrying with it the outer ends of the two connecting links. By this means the two lever arms are rotated together, equally, but in opposite directions. As these levers are connected to the two deflectors of the rudder by means of the rudder stocks, it is clear that rotation of the hand-wheel controls the opening and closing of the rudder and thus the speed and reversing of the boat. Steering is carried out merely by swinging the tiller as a whole exactly as with an ordinary tiller fitted to an ordinary rudder. When this is done the relative position of the two rudder deflectors has no tendency to alter since the whole arrangement is rigid in any position of the leading screw. This tiller gear is very easy to operate as we have found from practical trial. It requires no "learning" and the maneuvering, reversing and speed-control of a boat may be carried out easily by anyone who may only have seen the gear on first trying it.

In another form of this tiller gear, the opening and closing of the rudder—the speed control—is carried out by lifting or depressing the tiller, steering as before being carried out by swinging it. It is unnecessary to say more in detail of this arrangement, as it is clear that its design is merely one of the applications of suitable links and levers. The boat shown in Fig. 23 is fitted with this type of gear. Before leaving the question of the operating gear we must refer to Figs. 9 to 11, which show a tiller head gear for operating a Kitchen rudder of a heavier type and for a larger class of boat. An examination of these figures will show that the arrangement in effect is similar to the one already described. The gear in this latter case is intended to be operated by wire ropes, or chains, from a distance. As will be seen, the levers fitted to the heads of the rudder stocks are retained and are connected up by links to traversing nuts working on right and left-handed leading screws. These screws are rotated by gearing from the operating pulley, and it is clear that the rotation of this pulley traverses the nuts equally in opposite directions and swing the rudder as a whole to port or starboard. The opening and closing of the rudder to obtain speed control, and reversing is obtained by moving the leading screws with their gearing bodily as a whole. This clearly swings the rudder stock lever arms in opposite directions and opens or closes the rudder deflectors.

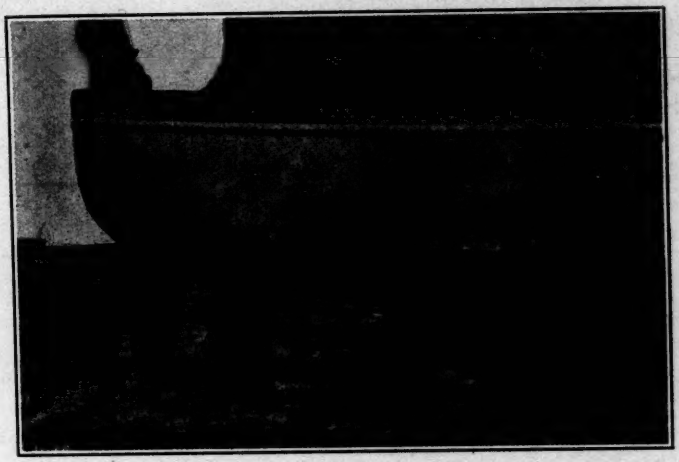


FIG. 21.

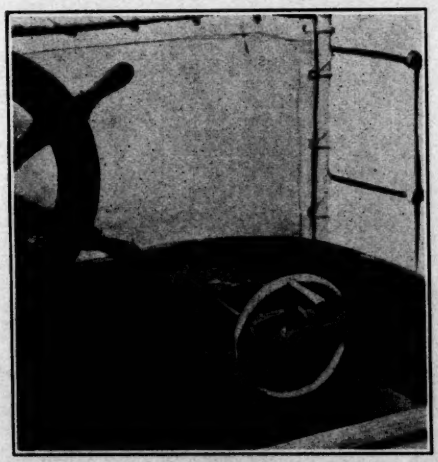


FIG. 22.

FIG. 23.

To enable this action to be carried out the leading screws and their gear are mounted on a sliding carriage which is able to move along the main bed plate. The position of this sliding carriage is determined by a rocking lever, which is operated by a third leading screw which can best be seen at the bottom of Fig. 11. This leading screw is moved to and fro by a nut carried by the chain or wire-rope operated control pulley. It will be seen that the two driving pulleys are fixed points, the pinion on the steering pulley spindle sliding along the spindle as the sliding carriage moves backwards and forwards. Fig. 22 shows a further type of arrangement, in which the control and steering wheels are mounted independent of each other.

Up to the present the Kitchen rudder has been fitted to a number of motor boats, Admiralty pinnaces, etc., driven by oil engines. Figs. 17 to 20 show the arrangement as fitted to a 50-foot, 150 b.h.p. Admiralty pinnace. It is in connection with such craft that the advantages of the rudder are most immediately apparent, in that it enables reversing gear to be dispensed with. The rudder is, of course, much simpler than a reversing clutch, feathering propeller, or other of the devices which have been used to enable a motor boat to reverse. Naturally to make place for itself the rudder must prove itself reliable, and in this direction it has a good record, no case of failure having occurred during its several years of trial. There is, of course, nothing in the construction of the rudder to make one anticipate failure. It is, however, not merely as a substitute for reversing clutches and similar devices that the introducers of the Kitchen rudder are hoping it may prove of service. The flexibility of control and rapid maneuvering qualities which the rudder gives are of importance in cases in which reversing in an ordinary way presents a difficulty. The control exercised by the rudder over the boat is so effective that a 25-foot motor boat can be turned on her own axis through 180 degrees in 30 seconds, while the reversing action is so powerful that its effect may almost be described as instantaneous.

We have had an opportunity of witnessing some trials of two motor boats fitted with the rudders, and were greatly struck with the maneuvering power exhibited. The steering control was very powerful and rapid in action, while the reversing power was remarkable. One of the boats, 25 feet long, running at 10 knots, was pulled up as though she had run into a mud bank. This rapid reversing is probably due to the fact that the propeller does not reverse, and consequently continues to operate in solid water. At the moment of reverse there was apparently no strain on the ship's structure. This is to be expected, as the conditions inside the boat do not change. Any strain which comes into play must be taken by the rudder, and this raises the question of the ability of the rudder to stand the work thrown on to it. Although the rudder may be moved over from full-speed ahead to full-speed astern as quickly as the helmsman is able to do it, there is none the less no sudden shock on the deflectors, which take a measurable time to close, so that the water is deflected gradually from a full stream astern to a full stream ahead. A further point is that the shape of the deflectors in itself gives considerable strength. The best reference on this question of mechanical strength of the rudder is, however, to practice, and during several years' experience with instant reversing and complicated maneuvering of every type there is no record of a rudder failure.

Records of various official trials of boats fitted with a Kitchen rudder are given below :—

Pinnace, 20-foot, 7 brake horsepower (Admiralty, 1919).

Ahead speed: 6.2 knots (same as with ordinary rudder).
Astern speed: 2.02 knots (propeller full speed ahead).
Full speed ahead to moving astern: 7 seconds.
Full speed ahead to dead stop: 4 seconds (in less than half boat's length).
Full speed astern to moving ahead: 4 seconds.
Diameter of turning circle at full speed ahead (helm hard over): About one length of boat.

Launch, 25-foot, 16 brake horsepower (Air Ministry, 1918).

Ahead speed: 9.80 knots.
Astern speed: 3.5 knots.
Full speed ahead to dead stop: Boat pulled up in 23 feet (four men aboard).
Full speed ahead to dead stop: Boat pulled up in 16 feet (one man aboard).
Time taken turning complete circle on own axis without progression in any direction: To starboard, 33 seconds; to port, 26 seconds.

Motor Boat, 35 Feet.

Full speed ahead: 7.6 knots (same as with ordinary rudder).
Full speed astern: 2.2 knots.
Full speed ahead to dead stop: 12 seconds (pulled up in 35 feet, its own length).
Full speed ahead to moving astern: 15 seconds.
Full speed astern to moving ahead: 6 seconds.
Diameter of turning circle at full speed ahead (helm hard over): 42 feet (time to complete circle, 43 seconds).

Admiralty Pinnace, 50 feet, 150 brake horsepower.

The boat with rudder full open was run over measured mile ahead and afterwards over part of same course with rudder closed. Astern speed was one-quarter to one-third of ahead speed. The astern speed, although not great, is quite enough for all practical purposes.

From full speed ahead to dead stop the boat was pulled up in 50 feet (her own length).

The stopping power of the boat is very much greater than the astern speed would indicate.

By closing the rudder to the neutral position the boat remained stationary (with engines running at full speed all the time). With the rudder in this position and the helm put hard over she described a circle about her own axis without moving ahead or astern.

Closing the rudder and giving her full helm the boat described a complete circle, the diameter of which was 75 feet, and the time taken 70 seconds and 74 seconds, port and starboard, respectively.

The time taken to reverse with this gear is very much less than with the ordinary reverse gear.

For maneuvering in and out of confined spaces the rudder is ideal, for you merely start up your engine, which has only a straight drive ahead, and do all your reversing and stopping by means of opening and closing the rudder.

It will be noticed that in some of the above records of trials the ahead speed is stated to be the same as that obtained with an ordinary rudder. This has been the case in all boats which have been equipped. We believe no boat has yet been built purposely to carry a Kitchen rudder. As the rudder can be made to conform to the streamline flow of the water aft, it is possible that such a boat might show a slight improvement in speed over a similar boat fitted with an ordinary rudder. The question of the fitting of the Kitchen rudder to a large class of boat is now under consideration. It is quite possible that even in steam vessels with ordinary reversing gear the speed and flexibility of maneuvering obtainable with the rudder will make it worth while fitting it for use during docking and similar operations. Nothing has yet been done, we think, in connection with twin-screw boats, but we understand that in such cases the rudder deflectors would be built to embrace both propellers. In conclusion, we should say that inquiries in reference to the rudder should at present be addressed to Mr. Gordon H. Fraser, of 747 Royal Liver Buildings, Liverpool. A company is, we understand, being formed to work the patent in the United Kingdom, and the rudder will shortly be available for all requirements.—"Engineering," May 16, 1919.

SHIP STRUCTURE IN RELATION TO EFFECT PRODUCED BY SUBMARINE WEAPONS.

1.—Trials lately conducted on the *Morosini* in Italy and on the *Jena* in France have led to the complete loss of the ship in both instances. This fact, which in the first case did not permit to ascertain the results of the explosion and in the second case must have rendered the examination less complete, cannot be said to encourage the repeating of tests of the same nature.

The significance of practical tests in the field of naval vessels architecture is very great; notwithstanding this, very little has been accomplished, due both to inherent difficulties and to expensive test. The result is that naval vessels are being built which have a structure not sanctioned by experience.

It seems to me, therefore, that the following of methods similar to those adopted for other architectural problems could eventually render less expensive and more apt to be controlled those trials made for the determination of the effects caused by the impact or by the explosion of projectiles or submarine weapons (mines, torpedoes, gymnotes) on the structures of ships.

Theoretically, the law of mechanical similitude can also be applied to the dynamical effects caused on a metallic body by the impact of solid or explosive projectiles and by the explosion of submarine charges. It can be said that: "If two mechanically similar structures are fired at with two similar projectiles shot at a similar initial speed and with similar explosive charges, it can be affirmed that the ratio of the deformations, whether elastic or not, caused by the explosion of the two projectiles are coördinated to the similitude of the two structures."

And also: "If two similar charges of explosives are made to explode in two mechanically similar places, the consequent deformations are similar."

Lastly: "If two similar guns must shoot similar projectiles at a similar speed, the charges, and above all the grain of the explosives, must be mechanically similar."

The demonstration of these assertions is a very easy matter. To verify the similitude law, let us consider what conditions must be met: First, the structures; second, the guns; third, the explosives.

First—If the ratio of the model to the ship is α and if E, R and K are the modulus of elasticity, the elongation and the safety load, the material of the model must possess the same elements respectively equal to:

$$e = \frac{E}{\alpha} \qquad r = \frac{R}{\alpha} \qquad k = \frac{K}{\alpha}$$

For instance, regarding high tensile strength steel, it may be assumed that

$$E = 2{,}400{,}000 \text{ Kg/cm}^2 ; \quad R = 6{,}000 \text{ Kg/cm}^2 ; \quad K = 1{,}200 \text{ Kg/cm}^2.$$

If the ratio $\alpha = 1/10$, the material must possess:

$$e = 240{,}000 \text{ Kg/cm}^2 ; \quad r = 600 \text{ Kg/cm}^2 ; \quad k = 120 \text{ Kg/cm}^2$$

It is not impossible to manufacture such a material; perhaps an alloy consisting of aluminum, lead and copper would answer to the purpose, although it would have a specific weight very near that of steel.

Second.—The similitude ratio of the testing guns and the structures must be the same; the projectiles must be shot with similar speeds, that is, if V is the real speed of the projectile, $\frac{V}{\sqrt{\alpha}}$ will be the initial speed of its model, while the ratio of the respective masses M and m is α^3.

If the tests concern only the structures and not the guns it is not necessary that the gun materials be similar. Therefore, the model of the gun may also be made of steel.

The velocity of a projectile is subordinate to the unitary pressure maintained in the bore during the passage from the breech to the muzzle. Now, if the ratio of the lengths of the guns is α, it would be sufficient for the ratio of the unitary pressures to be α anywhere in the bore to lead us to the conclusion that the ratio of the speeds of projectiles is $\sqrt{\alpha}$.

The unitary pressures are similar if the ratio of the weight of the charges is α^4 and if the grains are similar. If two charges of explosives, Q and q, are made to explode in two similar volumes, V_o and v_o, with the initial pressure P and p, the final pressures are

$$P' = \frac{P V_o + Q V'}{V_o} \qquad p' = \frac{p v_o + q V'}{v_o}$$

where V′ is the volume of gases caused by the explosion of 1 Kg of explosive at an ordinary pressure. If it must result that

$$P' = \alpha p'$$

$$\alpha P' = \frac{\alpha^4 p v_o + Q V'}{\alpha^3 v_o} \qquad Q = \alpha^4 q.$$

This result is not a contradiction of the similitude law because Q and q, are not similar weights, but weights which must produce similar volumes of gas at similar stresses.

The grains of the charges must also be similar. Senior Lieutenant Poma could assure me, basing his assertions on data obtained from

numerous firing trials, that in practice this law is very approximate. The charge of a 305 millimeter gun with initial speed of 780 m./s., is therefore 105 Kg. with the grain of the charge $9 \times 27 \times 600$ millimeters; the charge of a 30 millimeter gun, derived from the firing tables by interpolation, would result as 12 gr., while, according to the said law, should be 10.5 gr. with a similar grained charge.

Some trials made on this subject by our able artillery expert would lead to the determination of correction coefficients just as it is done at the Model Basin.

Third.—The ratio of explosive charges for the models of torpedoes, submarine mines, gymnotes, etc., as already said, must be α^4. In the case of gun cotton the grains have no influence whatever, even because it is a question of deflagration, while in the gun there happens a combustion.

Doubts could be entertained that the unitary pressures developed in the models are not similar to those of the ship, owing to the fact that the initial pressures are not similar, but are equal to the atmospheric pressure, it being not possible to maintain the model of say 1/10 at a pressure of 0.1 Kg/cm^2.

It is, however, easy to see that the differences of the compartments pressures and the external pressures are similar; the consequent deformations are therefore similar.

If the similar compartments are V_o and v_o-P and p, the final pressures

$$P = \frac{V_o + Q\,V'}{V_o} \qquad p = \frac{v_o + q\,V'}{v_o}$$

$$P - 1 = \frac{Q V'}{V_o} \qquad p - 1 = \frac{q\,V'}{v_o}$$

and as $Q = \alpha^4 q$, $P - 1 = \alpha(p - 1)$.

Fourth.—Elastic deformations caused by static loads are similar in similar structures. For instance, the elongation caused by the stress is expressed with:

$$A = \frac{N L}{E F} \qquad a = \frac{n l}{e f}$$

If $E = a\ e$. $\qquad A = a\ a$.

In the case of impacts of solid or liquid masses, as the kinetic energy of the striking mass must be identical to the deformation wrought, there exists the ratio of similitude between the first and second.

In effect, we have

$$\text{for the Ship M} \quad \frac{V^2}{2} = \frac{N^2 L}{2 E F} = \frac{M^2 L}{2 E I}$$

$$\text{for the Model m} \quad \frac{v^2}{2} = \frac{n^2 l}{2 e f} = \frac{m^2 l}{2 e i}$$

Lastly, the law of similitude concerns also injuries and deformations past the elongation limit, inasmuch as the fracture loads are similar. To verify the efficiency of a protection structure against submarine weapons and against projectiles, we could build a 1/10 model of the structure to be

tested, submitting it to the bursting of a submarine charge or the fire of a projectile. In this manner we would obtain, with great approximation, deformations similar to those which would result on the real structure if subjected to the shot of real guns fired with appropriate charges.

2.—The exact application of the law of mechanical similitude leads to the necessity of using a material similar to steel, that is, possessing an elongation modulus $E = 2400\ Kg/m.m^2$, a safety load $K = 1.2\ Kg/m.m.^2$ and a breaking load $R = 6\ Kg/m.m.^2$.

The few researches I could make on this subject have proved to me that it is difficult to find a metal meeting said characteristics. Besides, even admitting the possibility of producing the necessary alloy, the building of the models would entail a special working process, depriving the experiments of that simplicity and, above all, that economy which would have been their main assets.

By studying only the effects of the bursting of submarine charges on hulls of ships, I have strived to find the ratios that should exist between the charges and the dimensions, for the double purpose of: using for the model the same material of the real structure and to cause in both of them similar deformations.

We can now demonstrate that:

First.—On two similar structures, the deformations wrought by the bursting of two similar charges, the ratio of the weights of which Q and q would be α^3, are similar to the ratio α when the materials of both structures are identical.

Let two cylinders of the same material have a volume of V_o and v_o; a diameter of D and d; a height of H and h; a thickness of S and s, and having E as modulus of elongation and R as the breaking load. Let P be the initial unitary pressure. Let two explosive charges, Q and q, burst in the interior of the cylinders. Let

$$D = \alpha d \qquad H = \alpha h \qquad S = \alpha s \qquad V_o = \alpha^3 v_o$$
$$Q = \alpha^3 q$$

If V′ is the volume of gas at ordinary pressure produced by the bursting of 1 Kg of explosive, the pressures, after the explosion in the cylinders will be:

$$P' = \frac{P V_o + Q V'}{V_o} \qquad p' = \frac{P v_o + q V'}{v_o}$$

$$P' = p'$$

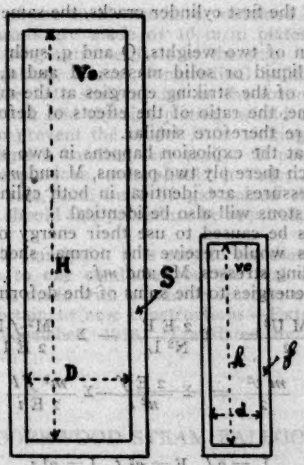
and the ratio of the elongations of the two cylinders will be α because:

$$A = \frac{N H}{E \pi D s} \qquad a = \frac{n h}{E \pi d s}$$

where N and n are the stresses caused by the fluid in the bottom of the cylinders:

$$N = P' \frac{\pi D^2}{4} \qquad n = P' \frac{\pi d^2}{4}$$

$$A = \frac{\alpha^3 n \cdot \alpha h}{E \cdot \pi \alpha^2 d s} = \alpha \frac{n h}{E \pi d s} = \alpha a$$



2. By analagous reasoning we could demonstrate that the permanent deformations must also be similar, because the tensile strengths are the same.

3. If a certain amount of explosive produces injury to some parts of said structure, the weight $q = \frac{Q}{a^3}$ will cause injury to a similar structure in the ratio α, provided the material of the two structures is the same.

Let R be the tensile strength of the material. In the case of the two similar cylinders, if the final pressure P′ causes the unitary pressure to reach the value of R in the first of them:

$$P' \frac{\pi D^2}{4} = R \pi D S$$

Even in the second cylinder the unitary tension will be the same to the breaking load:

$$P' \cdot \pi \frac{a^2 d^2}{4} = R \cdot \pi a^2 d s$$

$$P' \frac{\pi d^2}{4} = R \cdot \pi d s$$

In other words, if the first cylinder cracks, the same thing happens in the second.

4. If the explosion of two weights, Q and q, such that $Q = \alpha^3 . q$, causes the impact of two liquid or solid masses, M and n, against two similar structures, the ratio of the striking energies at the moment of the impact is α^3, and, being same, the ratio of the effects of deformation is the same. The deformations are therefore similar.

Let us suppose that the explosion happens in two similar cylinders with an open side, in which there ply two pistons, M and m. As after the explosion the unitary pressures are identical in both cylinders, the speeds imparted to the two pistons will also be identical.

If the two pistons be caused to use their energy on two similar structures, the structures would receive the normal shock stresses N and n and the normal bending stresses, Mf and mf.

By equalizing the energies to the sums of the deformation wrought:

$$\frac{M U^2}{2} = \Sigma \frac{2 E F}{N^2 L} - \Sigma \frac{M^2 f L}{2 E I}$$

$$\frac{m v^2}{2} = \Sigma \frac{2 E f}{n^2 l} - \Sigma \frac{m^2 f l}{2 E i}$$

and as:

$$L = \alpha l \quad F = \alpha^2 f \quad I = \alpha^4 i$$

we have:

$$N = \alpha^2 n \quad M_f = \alpha^3 . m_f$$

which means that the elongations:

$$A = \frac{N . L}{E . F} \qquad a = \frac{n . l}{e f}$$

and the inflections:

$$F = \frac{N L^3}{48 E I} \qquad f = \frac{n l^3}{48 E i}$$

are similar.

Being necessary to test the reliability of one or more structures systems, it is advisable to build the models in a convenient size, for instance, 1/10, limiting the structure to a length of 20 m., which would result in a 2 M Model.

As already said, the elastic and permanent deformations caused in the models by the explosion of submarine charges placed near the external plating will be similar to those that would be caused on a real structure by an explosive charge one thousand times greater.

This procedure possesses an absolute and relative value at the same time; relative because, if among the various structure models tested one resists the explosion of a certain charge better than the rest of the models, it is clear that the real structure corresponding to it will have the same advantages over the other structures; absolute, because after selecting the type of structure, it is possible to determine the weight of the charge capable of damaging it or the dimensions the structure should have to resist to the bursting of a determined charge.

Now that the difficulty pertaining to the material, which, as it was said, may be common steel, has been eliminated, no special devices or works are necessary to conduct a practical test.

A model is made of the 20 M structure contained between two frames. The terminal bulkheads are made of 10 m/m plates, because they must make up for the missing extreme parts of the ship.

The defensive planking, frames, longitudinal bulkheads, etc., would be built according to the 1/10 scale as far as the armored deck, which, together to the main deck and the side plating, would constitute the floating reserve necessary to prevent the model from sinking after the explosion.

A model of this kind is handled very easily. It permits the verifying of the effects of the explosions, even should it sink; might serve for two tests, one on each side, and, built of galvanized sheet and substituting welding to riveting, should this procedure be required, would not cost more than 500 lire.

From an economic standpoint it is easy to see that a test on a model is more advantageous to one conducted on the real structure: these tests could furthermore be made with a relative secrecy, which would be opportune if the tests pertain to new constructions.—Extracted from "Rivista Marittima," June-November, 1910. Translated by Lt.-Col. A. Guidoni, Royal Italian Navy.

THE LIDGERWOOD STEAM BALLOON WINCH.

The Lidgerwood Manufacturing Co. has constructed a special steam driven balloon winch and leading blocks for the handling of captive kite balloons on shipboard, with the object in view of reducing to a minimum the possibility of the balloon rope being broken due to sudden jerks.

CONSTRUCTION OF WINCH.

Duty.—The winch has a hauling-in pull of 5,000 pounds at a speed of 200 to 300 feet per minute, with 150 pounds steam pressure at the throttle. If the steam pressure is lower than 150 pounds the pull or speed will be reduced. The winch standing still will support a rope pull of 6,000 pounds.

Reciprocating Parts.—The winch is driven by means of two cylinders having 8¼-inch bore and 8-inch stroke, fitted with slide valves of the piston type, and will withstand a maximum steam pressure of 200 pounds. The connecting rods are drop forgings.

Control Valve.—Steam is admitted to the cylinders through a piston type steam reverse valve which is controlled by a single-hand lever. To wind in the rope, the lever is raised. To pay out the rope, the lever is depressed. This assumes that the rope winds on the top of drum. The winch cannot be reversed until the automatic brakes are unlocked.

Automatic Brakes.—Automatically operated band brakes lined with non-burn lining are fitted to each of the crank disks. The brake bands are made in halves, the halves being held together at the top by a cam and bolt device, having a handle. In the normal position this handle is down against the band, and the cam tightens the brake band.

When the handle is up, the brake bands are loosened, and the winch may be run in either direction. When the handle is down the brakes are locked and the winch may wind in the rope. Before reversing the winch to pay out the rope, these brakes must be unlocked.

Drum Shaft Assembly.—The drum shaft is driven through a single reduction of spur gearing from the crank shaft. A drum, 32 inches in diameter with 17-inch face, is bushed and allowed to run free on the shaft.

The drum is driven by two complete metallic slipping frictions, one upon each side of drum. These run dry and *no oil must be allowed to get on* their surfaces.

A set of composition friction blocks is bolted to the gear on one end and to a friction header (driven by a square portion of the shaft) at the other end of the drum. The other element of the slipping friction consists of two flat-faced steel rings, one bolted to each end of the drum, which engage the composition blocks. The friction is set up in the regular manner by cross-key and cross-key collar. A heavy spring is interposed between the cross-key collar and the friction header to compensate for the heat developed while the frictions are slipping. A winch head is keyed to the outboard end of the drum shaft for auxiliary purposes. *Do not allow the oil to get on friction surfaces.*

Automatic Friction Set Up.—A worm is turned on the drum flange. This worm drives a worm wheel which transmits its motion through bevel gearing to a sliding pinion which, when thrown in mesh with a gear fastened to the friction thrust screw, tends to set up the friction as the drum pays out the rope. *Do not throw the sliding pinion in mesh with hand wheel excepting when the winch is standing still.*

Rope Indicator.—A vertical graduated indicator is also driven from the worm on the drum flange and indicates the amount of rope paid out. In putting a 2,000-foot rope on the drum see that this indicator reads 2,000 feet when the winding of the rope is first commenced.

Tension Indicator.—A brass pin set in the hub of the friction handwheel serves as a tension indicator. When the friction is set up so that the end of this pin is flush with the hub of the wheel, the drum will resist a line pull of 3,000 pounds before the drum will slip. When the pin projects beyond the hub of the hand wheel, the friction is increased to amount indicated by graduations marked on the pin. This indicator is at best only approximately correct and should be corrected for wear.

LEAD BLOCKS.

There are two general methods of leading the rope to the balloon. On battleships, destroyers and ships which have a large after deck space free from masts, tall deck houses, etc., the rope leads through a special designed block direct to the balloon. This is placed near the stern post.

In the other case, on steamers such as merchantmen, patrol boats, yachts, etc., where there is generally a suitable straight spar, the rope leads from the balloon winch first to a deck block, located at the base of the mast and thence to a mast head block, from the top of which it leads away to the ballon. The object of this installation is to allow the balloon rope to lead in all directions at an angle of 45 degrees without fouling the mast, rigging, deckhouse, or other obstructions. With such installation, the hauldown block is used for the purpose of hauling in on the balloon rope to fetch the balloon basket in a suitable position to change observers.

Destroyer Type Deck Block.—This block is designed for the purpose of allowing the balloon rope to lead from it in any direction without danger of cutting or seizing the rope. To accomplish this, two sheaves are mounted in the same horizontal line in a frame composed of side pieces and brasses bolted to a swivelling bar. A bar bracket is mounted on each end of this bar for bolting the block to the deck. The block is then free to lean to either side as desired. Eye bolts are set in each side of block near the top, to which small ropes may be attached to prevent the block falling beyond 45 degrees to either side. The rope, as it comes from the winch, is led directly between two small wooden shoes placed in the rear

of the block to act as a brake, as will presently be described, and up between the two sheaves thence to the balloon. The oil cup on the block is provided with a wick which should be adjusted to fit against the tread of the sheave. This oil cup must be filled with a heavy oil, as heavy as will work through the wick.

The tread of the sheaves then being oiled, will in turn lubricate the rope. The wood brake blocks are for the purpose of putting a tension into the line (when required) as it leads to the winch. A slack rope will not coil firmly on the drum. Therefore, should the balloon rope become slack at any instant, these brake blocks may be sufficiently squeezed against the rope (by throwing the lever). The tension will then be set up in the rope as the winch hauls it in. The blocks should be entirely free from the rope at all times, unless such slack occurs. As the blocks wear they can be turned, presenting four faces for wear.

Leading Deck Block.—This block is designed with the swivel bearings, brake and oil cup features as previously described with the destroyer type block. The difference, however, lies in the fact that this block takes the lead from the winch, the rope then going (usually) vertically to the mast head block. A second sheave (a small one) is placed above the main sheave to insure the correct lead of the rope and prevent it from wearing against any guards, spreaders or side plates. Particular care should be taken in threading the rope through this block to be sure that it is threaded exactly. *This is important.* Any departure from the correct lead will invite a disaster.

Mast Head Block.—This block (originally designed for lashing to a mast) is to be placed at the highest possible point of the mast, and receives the lead of the balloon rope coming direct from the deck block. The sides of the block are bolted to a vertical swivelling bar set in bearing cast integral with heavy cast steel semi-bands. A special bearing is provided at the bottom bearing to take care of any downward thrust on the block. The block is held to the mast by lashings (to be used only in an emergency) passing around the semi-bands. Preferably these should be bolted or riveted to conventional mast bands. It is essential in this case also that the lead of the rope should be exactly as shown in the drawing.

Hauldown Block.—The hauldown block is a single sheave block having wood covered sides to keep from damaging the deck or anything it may strike against. In cases where a mast head block is used the hauldown block hangs on the balloon line between the mast head block and the balloon. Side pieces are extended downward for attaching the hauldown rope. These pieces are made of sufficient length so that their weight will cause the block always to hang upright on the rope. Any piece of manila or wire rope capable of withstanding a pull of 10,000 pounds may be used for a hauldown rope. This is led from the tail of the block to any snatch block (large enough to stand a pull of 10,000 pounds), on some open part of the deck suitable for landing the balloon basket. From this snatch block it is allowed to be led to any convenient drum or winch head which may be used for hauling in the rope. The snatch block and hauldown rope are not parts of the balloon winch equipment.

When the hauldown block is used in connection with the destroyer type deck block, it is hauled horizontally (rather than vertically) in the bight of the balloon cable. In bringing the balloon down, as the balloon basket nears the deck, it is pulled in board to the desired position by the hauldown block.

The bearing of all sheaves in all of the blocks are metalline bushed, thereby being self-lubricating, and should not be oiled, unless the metalline

bushing wears out and is replaced by some other type of bushing, because the oil will destroy the lubricating properties of the metalline. Oil holes with flush plugs are provided in the sides of the blocks to permit lubrication of the bushing should any other bushing be substituted for metalline.

SETTING UP OF WINCH AND BLOCKS.

The winch should be installed so that the rope led from the drum to the deck block (or nearest lead block) is at least 50 feet, preferably 60 feet, from the drum and exactly opposite the center of drum. This direct lead of 50 or 60 feet is necessary for the proper spooling of the rope on the drum. The lead of the rope is to come directly to the center line of the drum. The rope, in leading to the deck block, must lead parallel to the base of the block, and if coming directly from the drum, as is preferable, the rear end of the block must be shimmed to insure this correct lead.

The masthead block is to be placed at the highest possible point on the mast, so that there will be no danger of the balloon rope becoming entangled with any of the ship's fittings, no matter what the direction of the balloon may be. The balloon rope tends to swing around the mast head as in a 45 degree free cone, as it is impossible to tell what direction the balloon will fly relative to the ship.

When using the destroyer block, it shall be placed at the aftermost point of the deck suitable for such installation. This will tend to give the balloon a clear sweep aft. There must be a clear space above the deck represented by an inverted cone of 45 degrees.

Operation of the Winch.—The first winches constructed had the drum grooved for ¼-inch wire rope. The later winches had drum grooved for ⅜-inch rope. If the winch in question has ¼-inch grooving and ⅜-inch rope used, then it will be necessary to splice on to the end of the ⅜-inch rope 412 feet of ¼-inch rope. This ¼-inch rope will fill the groove in the drum, but should not be paid off from the drum. The ⅜-inch rope may then be wound on the succeeding layers in the regular manner. Not over 2,500 feet of ⅜-inch rope should ever be used on Mark I winch.

In starting to wind 2,000 feet of rope on the drum, the rope indicator is to be set at 2,000 feet. It will then register zero when the full 2,000 feet is wound on the drum; for shorter or longer lengths, the indicator must be properly adjusted. When ready to wind the rope on the drum, set up a slight amount of friction by turning the top of the friction handwheel over to the right. Be sure that the sliding pinion is *not* in mesh with the handwheel gear.

Raise the reverse valve lever, and the drum will turn in the proper direction for winding in the rope. When first winding the rope on the drum, it is essential that care be taken to coil this rope firmly and closely together. It will take between five and six layers to coil 2,000 feet of ⅜-inch rope. Should ¼-inch rope be used, 2,000 feet may be coiled in four layers. It is not safe to use much more than 2,500 feet of ⅜-inch rope with this size of drum, as the number of layers will become too great, and there will be danger of the oncoming rope getting wedged under previous coils and then snapping the rope when paying out at high speed.

SOCKET ON ROPE END.

Usually a socket or eye is spliced in the end of the rope for attachment to the balloon, and it will be necessary to lead the opposite end of the rope through the various blocks from the top down, before winding on the drum. It will frequently be necessary to remove the wooden brake blocks

in the deck block in order to permit any lashing on the end of the rope to pass through. An ordinary lashing, however, will readily reeve through all other parts of the blocks. If in reversing the winch to pay out the rope, the brake sticks and does not release readily, it is permissible to put a few drops of oil on the brake surface. After the brakes are released, the controlling throttle lever is lowered and the drum will pay out the rope. After the balloon is raised to required height the brakes are reset, so that the engine may not be overhauled and the friction is set up so that there will be a tension of 2,000 pounds in the rope before any slippage will take place. The control lever is then placed on center and pin inserted to hold it in place, thereby cutting off the steam, but holding it ready for immediate use.

The sliding pinion is then thrown in mesh with the friction handwheel gear. The balloon and winch may now be left to themselves. Should the ship speed up or the balloon encounter a heavy wind, either of which might tend to cause the stress in the rope to exceed 2,000 pounds, the balloon attempts to run away, the drum then starts to slip on its frictions and in so doing automatically sets the frictions up tighter until the running away of the balloon is automatically stopped.

The rope indicator shows the amount of rope that is paid out and conversely the amount that is left on the drum, and when the operator at any time finds the balloon too far away, he disengages the sliding pinions, turns on his steam and hauls the balloon in again to the proper distance.

If desired to bring the balloon down to change observers, steam is turned on and it is hauled down in the usual manner. As it nears the ship the hauldown block is put in operation and the basket of the balloon drawn down to a suitable point for the landing of the observer. After changing observers the hauldown line is first freed, thereby clearing the balloon from the boat. After that the automatic brakes are released and the balloon allowed to arise to the desired height.

DONT'S.

Don't ever have less than 50 feet direct lead from the center of the drum to the nearest leading block.

Don't ever turn steam on the winch while sliding pinion is in mesh with the gear.

Don't allow any oil to get in contact with the friction surfaces.

Don't forget to set the indicator at 2,000 feet before starting to coil rope on the drum.

Don't forget to release the brakes before trying to reverse the winch.

Don't start winch until the water is drained from cylinders.

Don't oil metalline bushings in the leading blocks.—Aircraft Technical Note, Navy Department.—"Aviation."

WARTIME NAVAL ENGINEERING.*

By Commander J. S. Evans, U.S.N.

Naval engineering comprises not only steam and electrical engineering, but also engineering work of the Bureau of Construction and Repair and the Bureau of Ordnance. However, I will limit myself in this paper to a discussion of the activities of the Bureau of Steam Engineering during the

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World War, from April 6, 1917, to November 11, 1918. This, on account of the fact that I am more familiar with the duties of this bureau than with those of the other bureaus.

ORGANIZATION AND DUTIES OF THE BUREAU OF STEAM ENGINEERING.

Before discussing the activities of the bureau, it is advisable to outline briefly its organization and functions. The basis of the bureau's organization is, generally, as laid down in Naval Regulations. These duties are divided broadly into the upkeep and repair of existing ships, and the building of new ones. This naturally divides into two parts, that is, the external work of the bureau and the internal work of the bureau. This external work consists of the coördination of this bureau's work with that of other bureaus and offices, with commandants and industrial managers of navy yards, commanders of fleets and ships in commission, inspectors of machinery, contractors for shipbuilding, inspectors of material, contractors for supplies, inventors and designers of improvements in mechanical and electrical apparatus and supplies; internally, the grouping of the related activities into divisions in the bureau, and the coördination of the work of these divisions and of individuals in each division.

The field covered by the bureau includes all that relates to the designing, building, fitting out, and repairing of machinery used for the propulsion of naval vessels, and this includes all steam and internal-combustion engines, boilers, pumps, heaters, distilling apparatus, refrigerating apparatus, all steam connections of the ships, the bulk of electrical apparatus for surface ships, submarines, and aircraft, machinery for aircraft, radio equipment, and the generation and supply of gas for lighter-than-air craft; also communication and submarine detecting devices. Part of the electrical apparatus on board ship is under the cognizance of the Bureau of Construction and Repair, this part being anchor handling gear, turret turning gear, steering gear, boat crane machinery, ventilation equipment, some drainage pumps, machine tool equipment of carpenter shop, and equipment of galleys and bake shops.

Other duties placed on the Bureau of Steam Engineering by the war were the repair and fitting out of the machinery installations of a number of converted merchant ships, and work entailed by the amalgamation of the Coast Guard and the Navy for war purposes.

SUB-DIVISIONS OF THE BUREAU.

The bureau is divided into several divisions, as follows:

Clerical,	Personnel and Fuel,
Design,	Logs and Records,
Electrical,	Inspection of Materials,
Radio,	Supplies,
Ship Repairs and Navy Yards,	Aeronautical.

The Chief of Bureau of Steam Engineering determines the bureau policy, authorizes all expenditures, and decides all important questions of detail. The Assistant Chief of Bureau handles routine business of a technical (except design) and administrative nature, supervises work and expenditures of machinery divisions of navy yards and repair bases, and condition of bureau work at private plants.

The Clerical Division is charged with the supervision of clerical, drafting, messenger and janitor forces of the bureau as provided by existing

regulations, the care, custody and operation of correspondence files, the supervision of matters connected with the expenditure of allotments, and, in general, all matter relating to the employment and promotion of classified employees for the bureau and its outlying offices as far as the bureau's appropriations are concerned.

DUTIES OF THE DESIGN DIVISION.

The Design Division is charged with the design of all machinery under the cognizance of the bureau, except such parts as are particularly assigned to the Electrical and Radio Divisions; the design, construction, installation, and test of boilers, engines, propellers, and all steam machinery under the cognizance of the bureau. It recommends the type of machinery to be adopted on new vessels, estimates the power required for the propulsion of such vessels, and makes the general arrangement design for the type of machinery adopted. In consultation with the Bureau of Construction and Repair, it determines the location and size of all machinery compartments, together with the arrangement of smoke pipes, uptakes, pipe passages, and the means of access to these; draws contract plans and specifications; investigates bids on alternative plans submitted by contractors and makes recommendations concerning the same; prepares directions for trials of machinery, and makes recommendations as to payments on contracts as far as meeting contract requirements is concerned. This division passes on all alterations of existing machinery under the cognizance of the bureau on all vessels. It also specifies materials and their characteristics for marine engineering construction, has supervision over the expenditures for machinery at various building yards and manufacturing plants, supervises the work of the Engineering Experiment Station at Annapolis, and the Fuel Oil Testing Plant at Philadelphia, furnishes data to the Supply Division for requisitions originated by the bureau, and to the Division of Inspection for specifications on engineering material, and makes recommendations on acceptance of important material when referred to this division by the Inspection Division; keeps record of percentage of completion of machinery on all ships building, and publishes same monthly; handles all matters in connection with changes of cost under contract; decides matters relative to extension of contract and claims for extension on account of delays.

THE ELECTRICAL DIVISION.

The Electrical Division is charged with the design of, and specifications for, all electrical apparatus under the cognizance of the bureau, except radio apparatus, and except main propelling machinery for electrically propelled vessels. Some of the items under the cognizance of this division are generators for power and lighting purposes, motors, controlling appliances, interior communication instruments, switch boards, lamps, search and signal lights; all power, light and interior communication wire and wiring appliances; also all mechanical signal systems under the cognizance of the bureau. Makes plans and specifications for complete electric plants of new ships other than electrical propelling machinery. In connection with the Bureau of Construction and Repair and the Bureaus of Navigation and Ordnance, has cognizance of the wiring plans of call bell systems, gyro gear, compass repeaters, and fire control instruments.

RADIO, REPAIR, PERSONNEL, AND FUEL DIVISIONS.

The Radio Division, in a similar manner to the above division, has cognizance of the design, manufacture, test, installation, maintenance, and repair of all radio outfits on board ships, aircraft, and at stations ashore, and submarine detection equipment.

The Repair Division considers all matters relative to repairs of vessels, and, if necessary, consults with the Design, Electrical, and Radio Divisions on matters for repairs pertaining to their duties, keeps the records of all engineering material and machinery in stock, and makes preparations for supplying merchant auxiliaries with necessary naval equipment on the outbreak of war.

The Personnel and Fuel Division draws up specifications for fuel for the fleet, makes recommendations as to award of contracts for fuel, and supervises its inspection. Also has charge of tests on fuel in connection with Design Division; makes recommendations in regard to engineering competition; investigates and makes reports on engineering subjects not directly connected with the duties of other divisions. Makes recommendations to Bureau of Navigation concerning engineering personnel.

LOGS, INSPECTION, SUPPLY, AND AERONAUTICS DIVISIONS.

The Division of Logs and Records prepares forms for use in the engineering department of vessels, receives, examines, and comments on engineering logs forwarded to the bureau, and prepares and issues to the service engineering information.

The Division of Inspection supervises the inspection of engineering material at plants of various manufacturers throughout the country, except at such stations or places where an inspector of machinery is stationed, and prepares Navy Department Leaflet Specifications for engineering materials.

The Supply Division handles requisitions for material, all matters relating to production and expediting delivery, analyzes bids submitted on requisitions and makes recommendations as to award of contract, after consulting with other divisions on the technical features of the apparatus called for on the requisition.

The Division of Aeronautics is charged with the design and supply of machinery and apparatus for aircraft, both heavier and lighter than air, equipment of aircraft stations so far as the bureau is concerned, and the supervision of the supply of helium.

During the war no changes in this organization were made, but the peace-time basis, which was inaugurated a number of years ago, was simply expanded, one item of the expansion being the increase of officers on duty in the bureau. On April 1, 1917, there were 35 officers on duty in the bureau, and on the date of the Armistice 143 officers.

MORE THAN 700 PER CENT INCREASE IN THE NAVY DURING THE WAR PERIOD.

At the time of the entry of the United States into the World War, the navy had about 350 vessels of all classes, built and building. When the Armistice came this had increased over 700 per cent, and the navy was operating and keeping in repair 2,634 vessels of all types and classes. Of these 783 were vessels of the regular naval service, battleship, cruisers, destroyers, gunboats, torpedoboats, mine ships, submarines, submarine chasers, and auxiliary ships. Ninety-three were vessels drawn from the Coast Guard, the Lighthouse Service, the Coast and Geodetic Survey, and the Fish Commission. One thousand four hundred and nineteen were

converted merchant vessels, comprising troop transports, mine sweepers, repair ships, tenders, hospital ships, and patrol vessels. Four hundred and thirty-two were vessels in the Naval Overseas Transportation Service. At the close of the war there were 350 naval vessels in service in European waters. There were under construction or contracted for 687 vessels of all classes, and 28 additional vessels had been authorized but not contracted for. Some of these vessels were being built by the various navy yards, but the majority of them were at private plants and were under the inspection of inspectors designated by the bureau. On the basis of displacement the 2,634 vessels on the navy list on November 1, 1918, had a displacement of 1,804,181 tons. The 627 being built or contracted for had a displacement of 1,247,494 tons, while the 28 authorized but not yet contracted for had a displacement of 153,000 tons.

ENLARGEMENT OF NAVY YARD FACILITIES.

The above gives a fair idea of the expansion of the naval service, and likewise of the volume of work necessary by the bureau to maintain vessels ready for active duty, to complete those building, and to complete the design of those authorized. In considering the volume of work, it must be remembered that a number of the vessels were old, that a number drawn from private sources were in a state not best fitted for naval uses and had to be converted and extensively overhauled to make them fit for such service. To handle the building and repair work of the machinery for this large increase, very extensive increase of facilities at the navy yards and certain private manufacturing plants were required. At all navy yards the plants of the Machinery Division were increased to a large extent, this being particularly the case at New York, Mare Island, and Puget Sound, where provision was made for handling work of the heaviest character, and at the Philadelphia, Boston, and Norfolk Navy Yards, where new machine shops and foundries were built. Eighteen bases were established in connection with the navy yards for the repair and maintenance of the machinery of gunboats, submarine chasers, submarines, and patrol vessels doing duty solely on the coast.

For repairs and supplies to vessels overseas, five major and two minor repair bases were established, three in France, one in Great Britain, one at Gibraltar, and two minor ones in the Mediterranean. Also the work of these repair bases was supplemented by six repair ships, which vessels did excellent work.

INDUSTRIAL REQUIREMENTS OF THE BUREAU.

In addition, a large amount of money was expended under the Bureau of Steam Engineering and other bureaus of the department to finance the extension of private plants or in building and equipping new ones to speed up the construction of machinery. An idea of the amount of this extension will be had from the fact that the following corporations had work entirely under the cognizance of the Bureau of Steam Engineering.

Chapman Valve Company,
Consolidated Safety Valve Company,
Edwards Valve Company,
Worthington Pump and Machinery Company,
Buffalo Naval Turbine Shop,
Providence Naval Boiler Shop,
De La Vergne Company,
Falk Company,
Sturtevant Company,
Griscom-Russell Company,
Allis-Chalmers Company,
Pollak Steel Company,
Camden Forge Company,
General Electric Company, Erie.

During the war the aeronautical work under the bureau jumped from practically nothing to huge proportions, as, for example, during the period covered about 8,000 aeronautical engines were ordered, of which about 2,600 were delivered.

The electrical work of the bureau increased proportionately with the increase in vessels, and, in addition to the general overhaul of electrical equipment of merchant ships taken over by the navy, the fire control systems of practically all fighting ships were remodelled and improved, and, on vessels drawn from outside sources, necessary signalling apparatus was installed.

In radio work the war placed on the bureau the work of outfitting with radio equipments all vessels taken over by the navy, and also on a large number of vessels of the Shipping Board and Railroad Administration, also all Government and privately owned radio stations were placed under the navy, and for the upkeep of these stations the Bureau of Steam Engineering was responsible.

NEW CONSTRUCTION.

Capital Ships.—Work on the regular navy building program was, during the war, subordinate to emergency construction. While work was continued on the construction of electric drive machinery for the battleships *Tennessee* and *California*, it was greatly curtailed on the machinery for other capital ships and was wholly suspended on the scout cruisers since, for the moment, their place could be taken by the destroyers. The battleships *Mississippi* and *New Mexico* were completed and commissioned, and work on a third, the *Idaho*, was carried out almost to completion when hostilities ceased. The main propelling machinery of the *Tennessee* and *California* was practically completed at the works of the manufacturers, but as construction of these vessels had been delayed by other work, it was not installed. Main propelling machinery (electric) of the type installed on the *New Mexico*, has been adopted as the most suitable machinery for capital ships, and is to be installed in four battleships (other than the *Tennessee* and *California*) of 30,000 s.h.p. each, six battleships of 60,000 s.h.p. each, and six battle cruisers of 180,000 s.h.p. each. Since this decision was reached during the war and the plans for the machinery of these vessels required a great deal of study and planning during the period covered, a short discussion of electric drive and reasons for its adoption in naval capital ships will be given later in this paper.

UNPARALLELED EMERGENCY CONSTRUCTION.

Destroyers.—In ships built and building, the United States has now a larger force of destroyers than that of any other navy. This increase has not been gradual, but has been obtained by giant strides within a year, and was made necessary by the appeal of the Allies for ships that could drive the submarine from the trade routes to England and the continent. The acuteness of this situation—lack of destroyers by the Allies at the time of our entrance into the war—has been told with much force by no less an authority than Admiral Sims in his articles that have appeared recently in magazines in this country—the details of which are no doubt familiar to all present. A destroyer is the acme of specialization in high powered, small space, light weight per horsepower engineering, and the difficulties encountered in the quantity production of these boats, which are modern in all respects, has no parallel in warship construction anywhere or at any time. On November 1, 1918, we had built or building 330 destroyers; of these 92

were in commission, and 238 were under construction with prospect of delivery of all within a few months. A number of these new destroyers were authorized August 29, 1916, 84 by act of March 4, 1917, 192 by act of October 6, 1917, 15 by act of July 1, 1918. Destroyers No. 75 to 94, inclusive, were contracted for before the war, and No. 95 to 185 were let in small groups. No. 186 to 347 were contracted for in large groups, and it was with these particularly that the greatest difficulties were met.

THE PROBLEM OF SUPPLIES.

When these contracts were awarded, destroyer construction was given precedence over all other work, but it was seen that new facilities for building would have to be created, especially as, in addition to the absence of available plants, there was a lack of enthusiasm among some of the shipbuilders.

Several conferences were held at the Navy Department between shipbuilders and representatives of firms manufacturing boilers, pumps, blowers, forgings, etc. When these conferences began the speed decided on up to that time was 28 knots. A subsequent change to 35 knots complicated the problem immeasurably. For example, it meant twice the boiler capacity, and with boiler tubes already becoming difficult to obtain, the problem as to tubes alone was one of considerable proportion.

Perhaps the greatest difficulty met with was in obtaining forgings. Twenty-eight knot speed allowed the use of solid forgings which could readily be obtained, but the 35 knot speed required hollow forgings, and for the manufacture of these there were only two large firms, neither of which was anxious for the work, as they preferred to make ordnance forgings. It was contemplated completing the destroyer program in eighteen months, and to do this it was clear that a large increase in the forging facilities of the country was necessary.

The shipyards were filled with work for both the navy and Emergency Fleet Corporation, but after several conferences with the shipbuilders it was decided to allot the first part of the October, 1917, program to the Newport News S. B. & D. D. Company (20 boats), New York Shipbuilding Company (20), Wm. Cramp & Sons Ship and Engine Building Company (25), and Bethlehem Shipbuilding Corporation (85), which last were to be built by the Fore River Plant and the West Coast Union Iron Works Plant.

ENLARGEMENT OF PRIVATE PLANTS.

To build the hulls it was necessary for Fore River to create a new shipyard at Squantum Point, near Boston. As the Fore River shops could take care of but a few sets of machinery of the 45 destroyers to be built at Squantum, it became necessary to build a shop for the manufacture of turbines at Buffalo and another for boilers at Providence, R. I.

At the Union Iron Works a new shop had been erected at Alameda to facilitate the construction of destroyers already under contract, and it was hoped that this shop would also be able to handle the machinery for the additional forty boats to be built there, but it finally became necessary to contract with the General Electric Company to convert their new shop, then building at Erie, Pa., to the special purpose of manufacturing turbines for the Union Iron Works' boats.

For the destroyers at Newport News S. B. and D. D. Company and New York Shipbuilding Company the turbines were contracted for with the

Westinghouse Company, who sublet the most important portion of the work to the Allis-Chalmers Company, of Milwaukee.

At Newport News Shipbuilding Company the situation was most difficult, but finally it was arranged to have the castings made at one place and the machining done in three others.

The boilers for the Fore River boats were built at the Naval Boiler Plant at Providence, those for Union Iron Works in San Francisco, and those for the other firms at the Babcock and Wilcox Company, where it was necessary to greatly enlarge their facilities.

ALLOCATION OF ENTIRE FACTORIES.

It was impossible to get pumps unless the Government undertook to enlarge existing plants, and Blake and Knowles Pump Works was the one decided on, and the works at East Cambridge, Mass., was almost entirely remodelled to take care of the pumps of all this latter program. Pumps were also obtained from the Warren Steam Pump Company and the Dow Pump Company.

Blowers were built by the Terry Turbine Company and the Sturtevant Company.

Evaporators and distillers were manufactured by the Griscom-Russell Company.

The foundry of the Wm. Cramp & Sons Ship and Engine Building Company was enlarged to produce the necessary propellers.

The works of the De La Vergne Machine Company, New York City, were bought outright to manufacture the turbines for the Cramp destroyers.

The turbine reduction gears were manufactured by the De Laval Steam Turbine Company, of Trenton, and by the Falk Company, of Milwaukee, the latter plant being enlarged to take care of this work, also by Westinghouse E. & M. Company, General Electric Company, Parsons, and Mare Island Navy Yard.

Other smaller establishments were equipped by the Navy Department to produce the necessary material; among them were the Edwards Valve Company, Chapman Valve Company, and the Wellman-Seaver Morgan Company, which last manufactured many of the condensers and much of the sheet metal work.

The cost of financing these industries has been estimated at thirty million dollars.

CONSTRUCTION OF SPECIAL TYPES.

Submarine Chasers.—The engines for this most useful type of vessel were of a standard tested design and 304 chasers were placed in commission before July 1, 1918. Standardization of design was carried out to the fullest extent, all being exact duplicates. These boats are 77 tons displacement, 110 feet long, 14 feet 8¾ inches beam, and 5 feet 5¼ inches draft. They are engined with three standard engines on three shafts, using gasoline as fuel. Type is four cycle six cylinders, cylinder diameter 10 inches, stroke 11 inches, and total horsepower of all three engines is 660. Their designed speed is 18 knots at 55-ton displacement. Fifty of these vessels were delivered to the French government and a large number of those operated by our navy have seen service abroad on the northern coast of Russia and in the Mediterranean.

Eagle Boats.—These interesting little boats, the building of which was undertaken by the Ford Motor Company, were modified destroyers. Their machinery requirements were unusual. First, it was essential that all machinery be noiseless and vibrationless; second, that it should be capable of being stopped instantaneously; and third, that it should be so designed as to permit of its rapid production on a "repeat" basis by inexperienced shipbuilders. The design adopted was most successful. Express type water tube boilers were specially designed, also an impulse turbine to drive the propeller shaft by planetary reduction gears. Rotary pumps were used for all purposes where this type was practicable. These plans were drawn up by the bureau in record time. These boats are of 500 ton displacement, length 200 feet 9 inches, draft 8 feet 6 inches, speed 18 knots. Oil fuel is used for the boilers. There are two boilers having a total h.s. of 3,000 square feet, and the horsepower of the main engines is 2,500—total weight of machinery 77.73 tons.

Submarines.—Duplicates of pre-war boats were built to meet the demand for emergency construction. The number of submarines increased from 37 built on January 1, 1916, to 87 built January 1, 1919, and 84 building.

Converted Merchant Vessels.—On twenty German and Austrian vessels fitted up as transports by the department, this bureau was required to install new refrigerating equipment, evaporators, distillers, and additional pumps. Three hundred and sixty-three vessels of the Naval Overseas Transportation service were similarly equipped. All of the vessels of the lake type required the installation of low main injections, as the main injection as fitted rolled out of water when the vessel went to sea.

REPAIR AND UPKEEP WORK.

During the whole war, although the navy was operating over 2,000 vessels, no one of importance was laid up permanently for repairs. The repair and maintenance of the 350 vessels in European waters was no simple matter. For this work the bases previously mentioned were established. Referring again to the six repair ships, which were provided, these proved to be of the greatest value to the fleet in the war area. Without the repair ships the destroyers could not have maintained the wonderful record they established of being always ready. We had an average of 50 destroyers serving nineteen months in European waters. One record case of work done by a repair ship was the retubing of the boilers of a destroyer in nine days without removing the boilers from the ship. The destroyers were given a periodic five-day overhaul period, and the strict observance of this rule had much to do with the success of the destroyers. For example, the boilers of one destroyer needed retubing, and this was done without interruption of her convoy and patrol duty by completing one side of a boiler on one repair period, the other being retubed on the following period.

The efficiency of the machinery divisions at the various navy yards and shore stations had a most important effect on the general success of the bureau's work in keeping ready for service the machinery of the fleet, and of other vessels operated by the navy. An example of the amount of work carried on is that of the Navy Yard, New York, which had charge of the conversion and repair of 723 vessels of all classes, the majority of the work being under the Bureau of Steam Engineering.

REPAIRS ON THE INTERNED GERMAN SHIPS.

Perhaps the most interesting item of repair work done by the bureau was the repair of the ex-German ships damaged by the vandalism of their crews. When the United States entered the war there were in all 103 German ships seized by our Government, of this number about 50 were turned over to the navy by the Shipping Board for repairs. The major damage inflicted was the breaking of cast-iron parts of the main engines, chiefly the cylinders. In addition, boilers were burned out by dry firing, boiler stays, connecting rods and piston rods were sawed through, and many minor items of damage were also found. In considering this matter it must be borne in mind that the engines of these ships had not only been damaged, but that normally they should have been in poor condition, since the ships had been lying inactive in various American ports for three years, with very inadequate upkeep during that period. Hence, deterioration as well as damage had to be corrected, and no drawings or plans of any kind were available. The steps taken to repair the vandalism were so effective that less time was consumed in making these repairs than in the general overhaul necessary.

The chief problem that confronted the bureau was the repair of the broken cylinders of the reciprocating engines. It was estimated that it would take eighteen months to replace these cylinders with new ones. On the report of Captain E. P. Jessop, then Engineer Officer of the New York Navy Yard, supported by Admiral G. E. Burd, the Industrial Manager of that yard, the bureau decided, and immediately issued orders, to make repairs where possible by electric or oxyacetylene welding, and when welding was impossible to resort to mechanical patching, that is, to use "soft patches" secured by body bound bolts. A few of the large cylinders were repaired by oxyacetylene welding. However, on account of the necessity of removing the cylinder from the engine this method was abandoned and electric welding resorted to. Much criticism was leveled at the bureau for using these welding methods by not a few engineers and shipbuilders, but the results obtained have justified the procedure in every way. The vessels were put in service a year before the hostilities ceased, and have operated without a mishap, due to the repair methods employed.

RESEARCH AND DEVELOPMENT OF A NUMBER OF SPECIAL ITEMS.

Radio telegraphy was developed during the war by the Bureau of Steam Engineering, in coöperation with civilian organizations to a truly remarkable degree. Some items of this advance are as follows:

1. The increase made in efficiency of trans-oceanic radio service has made it a serious competitor of the cable. Five stations have been developed on the Atlantic Coast capable of continuous communication with Europe. The station at Annapolis, built during the war, is the most powerful in the United States.

2. For trans-oceanic reception photographic recorders were developed which permit of the reception of forty words per minute.

3. Sleet melting equipment for antennæ was developed and installed.

4. The development and installation of a high frequency alternator and the adaption of this equipment to permit its use as long distance radio telephone transmitter.

Example: The *George Washington*, in harbor at Brest, received radio telephone messages from New Brunswick, N. J., a distance of 3,200 miles, and by use of vacuum tube telephone was able to reply when still 1,300 miles at sea.

5. Development of underwater radio equipment for submarines whereby messages can be received by the submarine when submerged 20 feet.

6. Development of the radio compass to assist navigation. Vessels off port in a fog and unable to enter have been successfully guided to the light vessel and thence to the channel entrances by shore radio compass stations. Destroyers unable otherwise to regain contact with convoys were able to do so by steaming on courses as shown by the bearing of the convoy from the destroyers' radio compass. Radio compasses for aircraft were developed such that bearings could be taken within one degree from a station at a distance of 300 miles.

7. Radio equipment for aircraft has been developed to a remarkable degree, the highest power set being capable of telephonic communication at 200 miles and by telegraph 600 miles. During the trans-Atlantic flight of the N.C. boats messages were received by the boats at 1,600 miles and communication was maintained with shore up to 700 miles.

LISTENING DEVICES.

During the early stages of the war, when the enemy submarines were most active and successful in the destruction of shipping, it was recognized that anti-submarine vessels were not largely effective, due to the fact that the location of a submerged submarine could not be determined. An immediate solution of the problem and the expeditious development and manufacture of devices for submarine detection were a vital necessity.

All conceivable methods whereby submerged submarines could be detected and located were experimented with. Eight devices have been developed under the bureau and made in quantity not only for our own navy, but also for the British and French navies. Also modified forms of these devices have been designed for use in the protection of the Atlantic Coast of the United States. The listening devices developed in this country are considered the most effective devices in existence, for the purpose. Before efficient listening devices were developed a submarine, when submerged, was practically immune from attack. After the perfection of the instruments, entirely different conditions prevailed, and a submarine underway, either on the surface or submerged, was never safe from detection and attack. The activities of the enemy submarines were greatly curtailed. When the German submarine commanders learned, through bitter experience, that as long as their motors were running, the location of their boats could be ascertained by any nearby vessel equipped with the detection apparatus, they ceased taking hazardous chances. Instead of going when they pleased and making surprise attacks at will, they found it meant almost certain destruction to approach the vicinity of enemy vessels.

ELECTRIC PROPELLING MACHINERY.

Referring again to the adoption of electric propelling machinery for capital ships, the fact that a very large part of its development was carried on during the war period makes it of special interest. The characteristics of this type of propulsion which make it so suitable for capital ships have already been set forth in various papers, and but a brief synopsis of such characteristics will be given now. These are (*a*) ease of control (*b*) increased economies at cruising speed, and (*c*) its suitability for installation of turbines and generators near the boilers (eliminating long steam lines), motors where best suited, and for a general rearrangement of the machinery spaces over that of the older ships, so that the naval constructor is much less hampered in his efforts to secure maximum

protection of all vital parts of the ship. The machinery for the *New Mexico* was completed, tested, and installed during the war period. The vessel was commissioned, and after dock trials the vessel went to sea, also during this period. The fact that the bureau and the department had the utmost confidence in the type of machinery installed is shown by the fact that on her first trip to sea the vessel was directed to maintain a standard speed of not less than 17 knots. This was accomplished with ease, as was also the next trip at higher speed. The vessel has been an unqualified success. However, even before the tests of this ship were completed, the same type of machinery was recommended and adopted for the new vessels. I believe that all present will agree that the bureau must have done an enormous amount of work in this connection to arrive at a decision of such importance during such a period.

TRAINING THE ENLISTED PERSONNEL.

Although, strictly speaking, the training of enlisted personnel may not be considered as an engineering matter, I believe some mention should be made of it, for the development of a personnel to handle the engineering equipment of the vessels afloat was a serious problem. But such a personnel was developed in the fleet for transfer to vessels where needed, and the fact that, as before stated, during the war no vessel was permanently laid up for repairs, is, in my opinion, evidence that the work of the bureau and the work of the engineering personnel on board ship was work well done.—"The Journal of the Engineers' Club of Philadelphia."

H.M.S. "*HOOD*."*

By Sir Eustace D'Eyncourt, K.C.B., Vice-President.

Following on the paper I read before this Institution last year upon Naval Construction during the war, in which I gave a brief account of all the principal vessels designed and constructed during the war period, I now propose to give an account of H.M.S. *Hood*, which will complete the series of big ships designed and built for the navy down to the present time.

Apart from the smaller craft, such as destroyers and patrol boats, of which an immense number were required during the war, the type of large vessel in which we were somewhat deficient in 1915 was the battle cruiser. It was known that the Germans were building several very important ships of this type, including the *Derfflinger*, *Lützow*, *Hindenburg*, and others to follow. Our own vessels were later to be supplemented by the *Renown* and *Repulse*, and there were the very large light cruisers *Courageous*, *Glorious* and *Furious*, but these three latter could not be regarded as battle cruisers in the proper sense of the term.

* Paper read at the spring meeting of the 61st session of the Institution of Naval Architects, March 24.

TABLE.—*Leading particulars of H.M.S. "Hood."*

Length between perpendiculars	810 ft.
Length, overall	860 ft.
Breadth, extreme	104 ft.
Load draught (L.W.L.), mean	28 ft. 6 in.
Displacement at load draught	41,200 tons.
Shaft horsepower of engines	144,000
Speed at load draught	31 knots.
Oil fuel at load draught	1,200 tons.
Oil fuel capacity	4,000 tons.
Armament	8 15-in. B.L. 12 5.5-in. B.L. 4 4-in. H.A. 2 21-in. T.T. submerged. 2 double 21-in. T.T. above water.
Armor—	
Side—Amidships	12 in., 7 in., 5 in.
Side—Forward	6 in., 5 in.
Side—Aft	6 in.
Bulkheads, forward and aft	5 in., 4 in.
Barbettes	12 in.
Gunhouses	15 in., 11 in.
Conning-tower	11 in., 9 in.
Protection—	
Vertical plating, magazines, etc.	1½ in.
Forecastle deck, amidships	2 in.
Upper deck, amidships	1 in.
Main deck	1½ in. flat / 2 in. slope increased to 3 in. over magazines.
Lower deck forward	1 in., 1½ in.
Lower deck aft	1 in., 3 in.

Weights expressed as percentages of displacement at load draught:—

Item.	Per cent.
Equipment	2.0
Armament	13.5
Machinery	13.0
Oil fuel	3.0
Armor and protection†	33.5
Hull†	36.0
	100.0

I therefore received instructions in 1915 to prepare designs of battle cruisers, embodying the latest ideas in regard to underwater protection, speed, etc. A considerable number of designs were submitted to the Board, but as there were no large berths available, it was not possible to place an order before the spring of 1916, and in March of that year the board selected a design of the dimensions given here to be worked out in detail and proceeded with. This design formed the original basis for the design of the *Hood*, having the same length and breadth as the *Hood*, but

† The plating of the decks and side is required to be of such thickness for strength that it contributes very materially to protection. The weight of this plating is, however, included in "Hull."

of considerably less draught, viz., 25 feet 6 inches normal and 29 feet deep, with a displacement of 36,300 tons on a length of 810 feet between perpendiculars. The speed was to be 32 knots, with machinery of 144,000 s.h.p. The armor, which was approximately equivalent to that of *Tiger*, consisted of an 8-inch belt and 9-inch barbettes. The armament was to be eight 15-inch guns and sixteen 5½-inch guns, together with two 21-inch submerged torpedo tubes.

I strongly advocated the adoption in this design of small-tube boilers, and these were accordingly included for the first time in the design of a capital ship. This feature had the most marked influence on the whole design, on account both of the smaller space occupied and the reduced weight involved, as compared with the space and weight which would be necessary for the large-tube boilers which had hitherto been adopted in all our big ships, including *Renown* and *Repulse*. The first big ships, though they cannot be regarded as capital ships, as mentioned above, which had the small-tube boilers were the *Courageous*, *Glorious* and *Furious*.

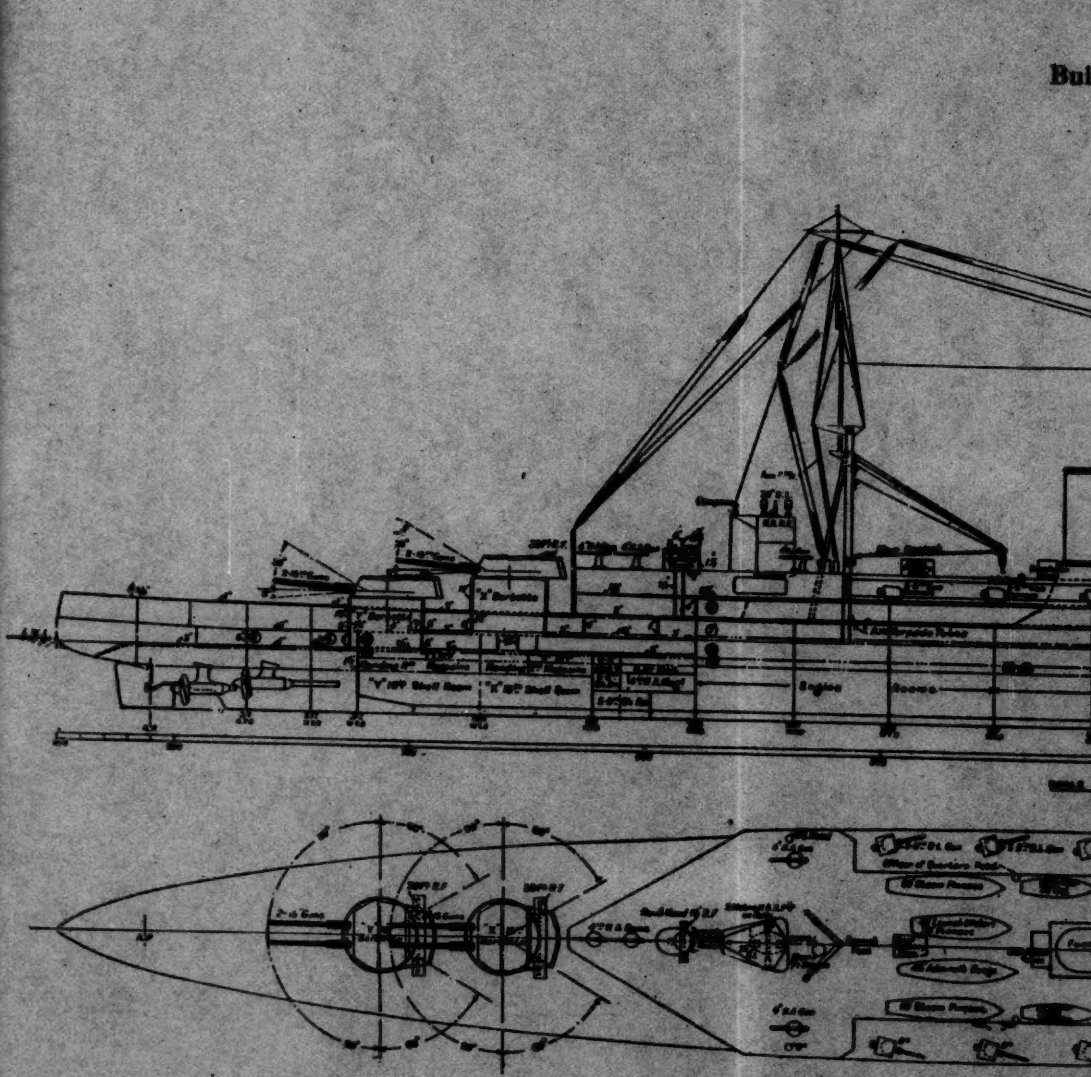
The original design of *Hood* was approved by the board in April, 1916, and orders were placed to build one ship each with John Brown & Co., Limited, Cammell Laird & Co., Limited, and the Fairfield Shipbuilding & Engineering Company, Limited, the ships being named, respectively, *Hood*, *Howe*, and *Rodney*. Subsequently an order for a fourth ship, H.M.S. *Anson*, was placed with Sir W. G. Armstrong, Whitworth & Co., Limited.

INFLUENCE OF THE BATTLE OF JUTLAND.

It will be remembered that the Battle of Jutland took place on May 31, 1916. This, the greatest fleet engagement of modern times, naturally led to further consideration of the design of the *Hood*, which had only just been ordered; and, in view of the damage which was done to our own battle cruisers and also to the German ships of similar type, it was deemed advisable to increase the armor protection, if possible. As the result of very extensive investigations, it was found possible, by accepting a deeper draught and a slightly reduced speed, to add very considerably to the protection of the vessels as already designed, without serious modification to the design of the ship as a whole. Accordingly, in September, 1916, definite proposals for increased protection were submitted. The alterations were of a very radical character, the armor belt being increased from 8 inches to 12 inches, and the barbettes from 9 inches to 12 inches; and certain increases were also made in the deck protection. The particulars, as thus finally decided, are given in the table. At the same time the eight 15-inch gun mountings had their design modified to admit of an elevation of 30 degrees, and certain other modifications were made, both in the torpedo armament and also in the arrangements, for preventing the flash penetrating to the magazines—a form of protection which was elaborated for all our ships at this time. All these increases involved an additional weight of nearly 5,000 tons, the legend displacement of the *Hood* becoming finally 41,200 tons when carrying 1,200 tons of fuel; the ship then having a draught of 28½ feet, and a draught of 31½ feet with full fuel load, viz., 4,000 tons.

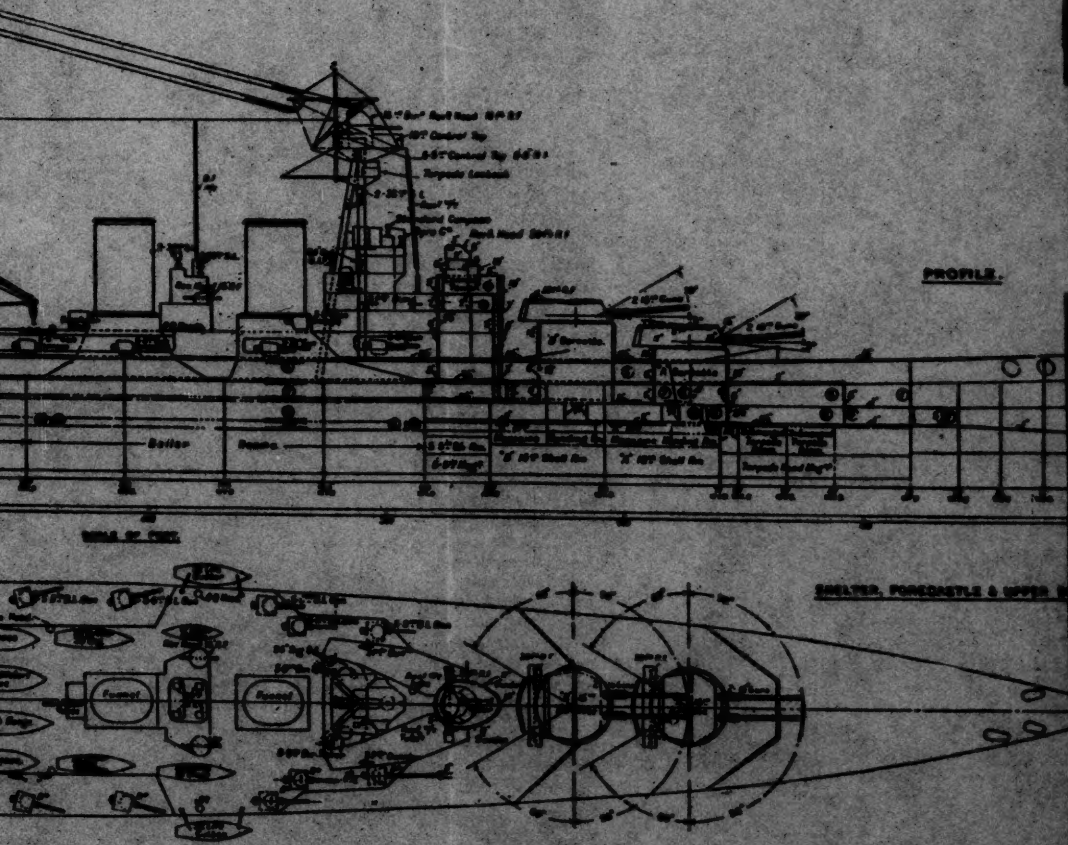
The original length and beam of the ship were maintained as before. Some extra plating had to be provided on the decks for strength purposes, but the underwater protection against torpedoes, which had proved very efficient, was retained, as in the original design.

I should mention also that the stability conditions were such that the increased displacement could be accepted whilst still maintaining a very satisfactory metacentric height.



H.M.S. "HOOD."

Built and Engined by John Brown & Co., Ltd., Clydebank.



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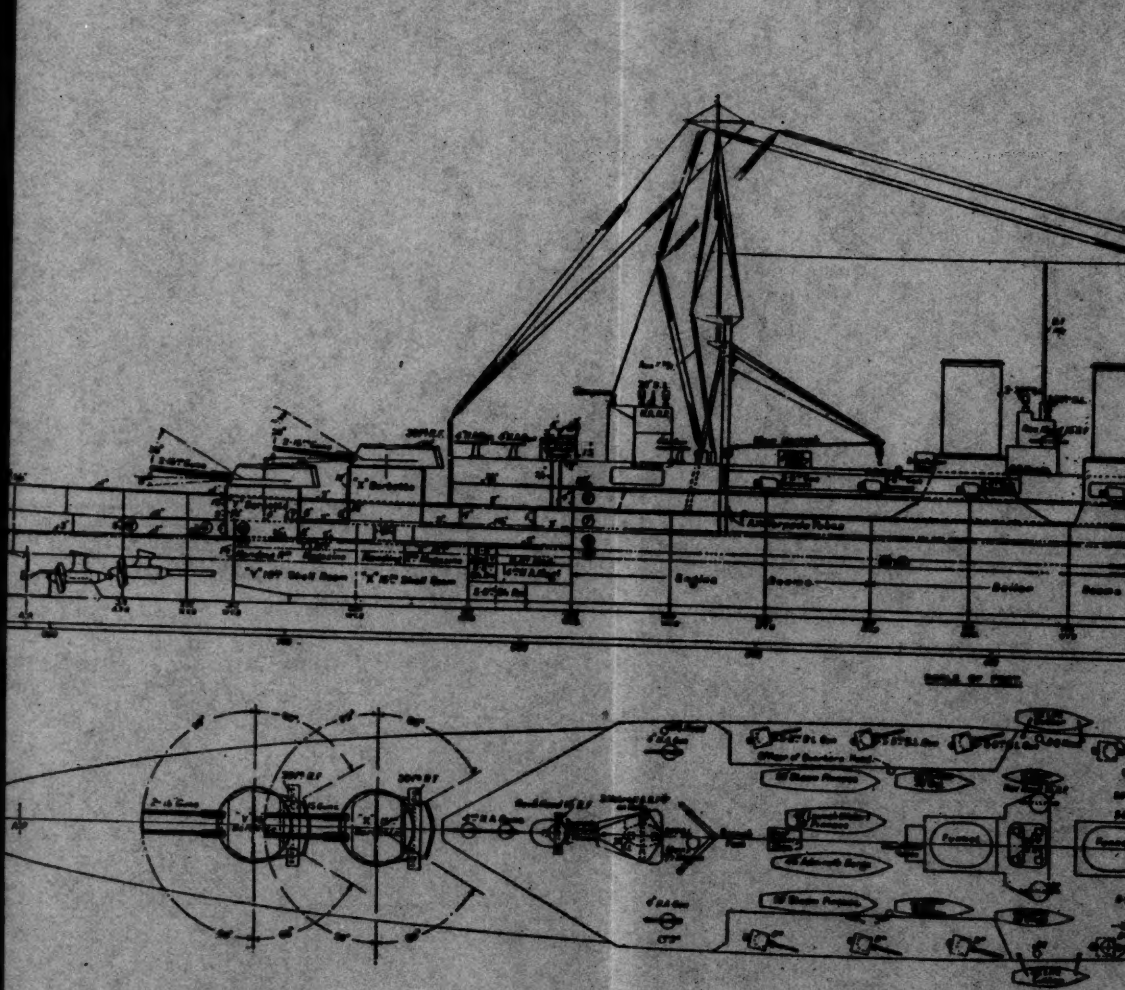
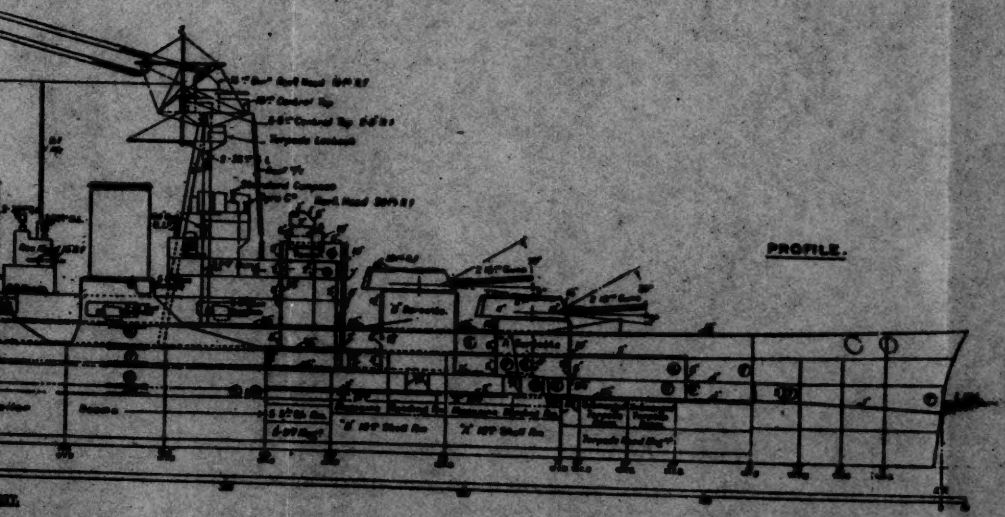


H.M.S. "HOOD."

and Engined by John Brown & Co., Ltd., Clydebank.



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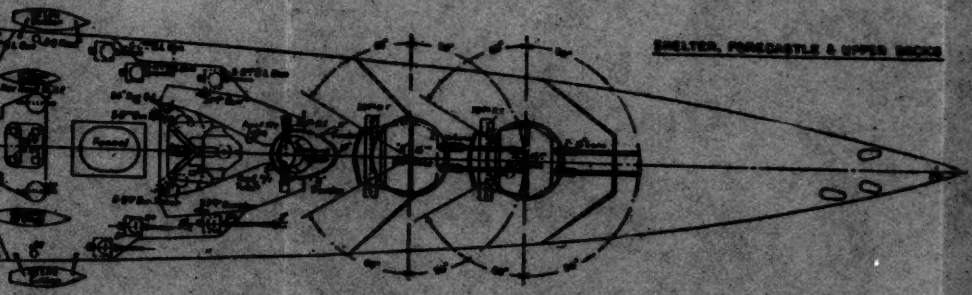
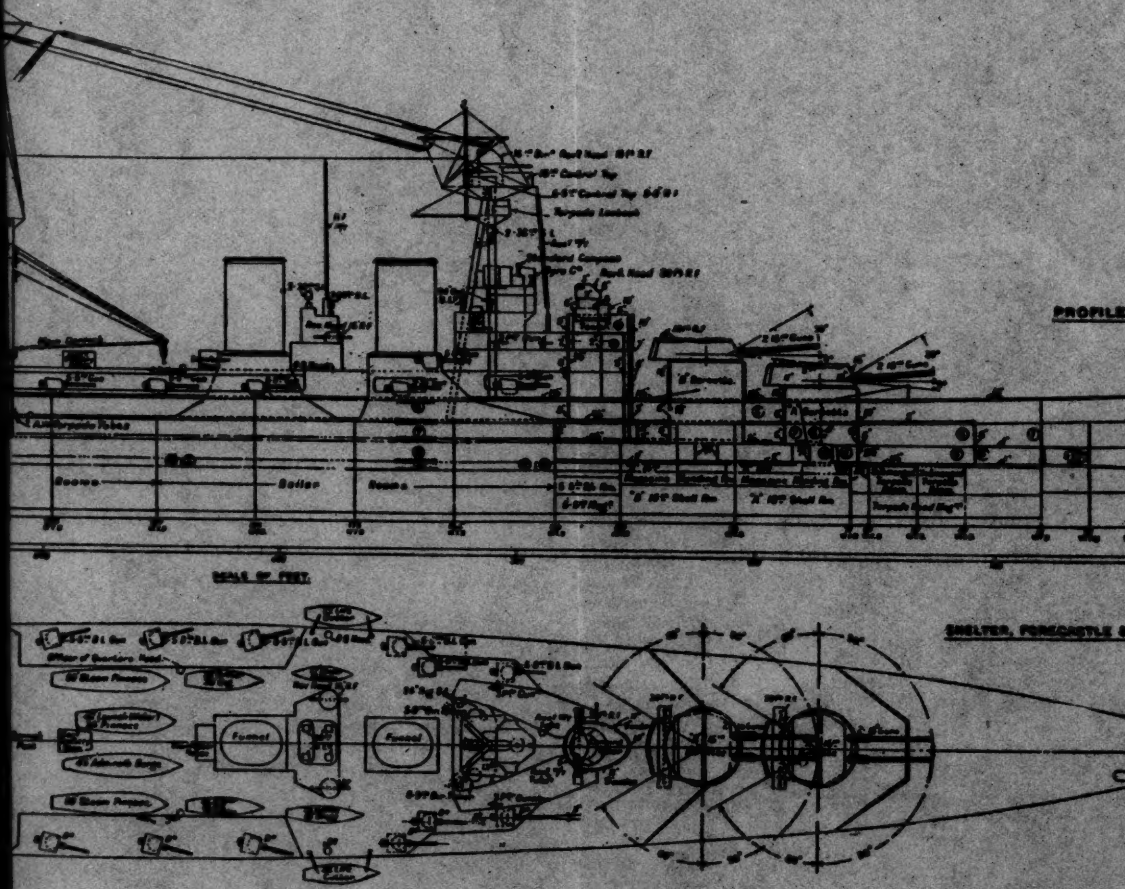


SHELTER, FORECASTLE & UPPER DECKS

H.M.S. "HOOD."

Built and Engined by John Brown & Co., Ltd., Clydebank.



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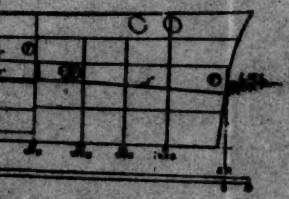
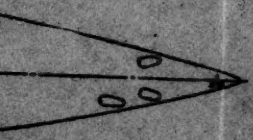
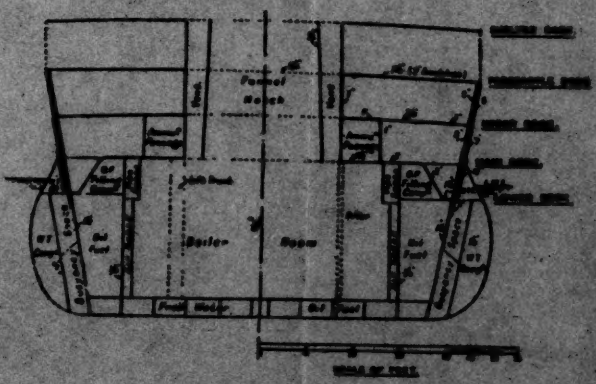


FORECASTLE & UPPER DECKS



SECTION THROUGH MIDDLE BOILER ROOM.



DESIGN FINALLY APPROVED IN 1917.

All the modifications made in the design after the Battle of Jutland were considered in consultation, not only with the Board of Admiralty, but with the Commander-in-Chief, involving, as they did, special arrangements for the fire and torpedo control, arrangements of bridges, etc.; and it was not till 1917 that the design was finally approved in all its details. This change naturally militated against the quick construction of the ship, and it will be about four years from the approval of the original design, in April, 1916, to the time of her completion, this being about double the time taken to build our recent capital ships, and nearly three times that taken to build H.M.S. *Repulse* and *Renown*. It is only fair, however, to say that the modifications were quite justified by the circumstances, and they have no doubt made the ship a much more powerfully-protected one, whilst increasing her displacement, and consequently the weight of material to be worked, to about 50 per cent more than that of *Repulse* and *Renown*.

144,000 S.H.P.

The main machinery, consisting of geared turbines to develop 144,000 horsepower, is the largest power which has ever been put through gearing, namely, 36,000 horsepower on each of the four shafts. The machinery is placed in three engine-rooms, of which the forward one contains two independent sets for the outer shafts; the middle and after engine-rooms containing one independent set for each of the inner shafts. This power, which was designed to give 32 knots for the earlier design of 36,300 tons displacement, is expected to give at least 31 knots in deep water at a displacement of 41,200 tons for the *Hood* as built.

The revolutions per minute of the propellers are 210 at full speed, and this admits of the adoption of propellers of higher efficiency than many we have had in our capital ships before gearing was introduced, when the revolutions were considerably higher.

The boilers, 24 in number, as mentioned above, are of the small-tube type with forced draught, and are arranged in four boiler-rooms. It was known that a great number of the later German capital ships took advantage of the reduced weight and space occupied by small-tube boilers, and have generally adopted them in their recent designs. It is hardly necessary to say that oil is the only fuel used in *Hood*.

MAIN ARMAMENT.

It will be seen from the plate drawing that the main armament of eight 15-inch guns is mounted, as in our recent battleships of the *Queen Elisabeth* and *Royal Sovereign* classes, in four turrets, all on the center line. The heights of the axes of the guns above the normal load waterline for each turret, commencing from forward, are 32 feet, 42 feet, 31 feet 9 inches and 21 feet 9 inches. The big guns have very large arcs of training, the forward ones training to 60 degrees abaft the beam, and the after ones to 60 degrees before the beam.

The anti-torpedo boat destroyer armament consists of twelve 5½-inch guns with shields 1 inch thick; these guns being arranged on the forecastle deck and shelter deck as shown. There are also four 4-inch anti-aircraft guns on the shelter deck aft.

There are two 21-inch submerged torpedo tubes, each in a separate compartment forward, and four 21-inch above-water torpedo tubes between the upper and forecastle decks; these above-water tubes being a further addition since the original design was made.

The distribution of armor is shown on the plate drawing. The 12-inch belt has a length of 562 feet and a depth of 9 feet 6 inches. Above the main belt is a strake of 7-inch armor to the height of the upper deck, and above that again there is 5-inch armor between the upper and forecastle decks. The side armor all slopes outward from below, as seen on the section, so that the virtual thicknesses are really somewhat greater, as the shell cannot thus hit the armor normally. There is thick plating behind all the armor, varying from 2 inches over the greater portion to 1¼ inches and 1 inch elsewhere.

TURRET PROTECTION.

The gun-shields for the turrets are of improved shape, with rather flatter roof than formerly, and the armor has a thickness of 15 inches in front, 12 inches and 11 inches on the side, with a thick roof plate. The conning tower arrangements have been very specially considered, and arrangements made, in addition to the Admiral's conning-tower, for torpedo and 15-inch gun control towers, and 5½-inch gun-control, revolving hood with range-finder, in addition to the range-finders provided in each turret. There are also smaller range-finders in connection with the different controls.

The torpedo protection consists of the bulge arrangement, with an outer compartment of air and an inner one specially strengthened with the necessary separating bulkheads, etc. This protection extends throughout the whole length of the machinery spaces and magazines, and it can be said that it renders the ship as safe against attack from torpedoes under water as she is against gun attack above water.

The oil-fuel tanks are arranged along the sides, thus giving additional protection.

It is a somewhat curious coincidence that the first series of recent experiments on under-water protection should have begun by a set of trials of torpedo charges against the old *Hood*, of the old *Royal Sovereign* class, built in 1893. Developing the lessons learnt from these trials, which took place in the years immediately preceding the war, the bulge protection was first designed and fitted at the commencement of the war in the old cruisers of the *Edgar* type, to which I alluded in my last year's paper. A series of experiments were then made at the request of the Admiralty by the late Professor (afterwards Colonel) Bertram Hopkinson, F.R.S., in conjunction with myself, at first under the auspices of the Royal Society. These experiments, which were made on different scales gradually working up to the full size, were of the greatest value, and the original protection fitted to the *Edgar* was modified and the details of construction as modified have been largely adopted in the *Hood*. I am glad to have this opportunity of paying special tribute to the work of Professor Hopkinson and to his memory.

THE VALUE OF THE TORPEDO "BULGE."

It may be noted that no ship provided with the original or the later form of bulge was lost, nor even seriously damaged by torpedo attack during the war, nor was there loss of life in any case; this being chiefly due to the special form of bulge and to its being external to the ship proper. By careful attention to the form the reduction in speed is only that due to the added displacement, amounting in the *Hood* to only a fraction of a knot.

Experience and trials have shown that the *Hood*, as protected, can receive the blows of several torpedoes and still remain in the line without serious loss of speed.

DOCKING QUESTION.

In the design of the *Hood* special consideration had to be given to docking, and she is provided with double internal vertical keels along the center line, and the docking keels come under the longitudinal bulkheads to support the ship on each side. Arrangements are made to dock the *Hood* in our larger docks, and she has already been docked and prepared for trials at H.M. Dockyard, Rosyth.

I should like to add that it was never proposed to dock the *Hood* at Devonport, although one of the technical papers stated recently that it was discovered that she could not be docked there. Whenever a design is got out at the Admiralty, or presumably anywhere else, when fixing the dimensions, one of the first points to consider is, where the ship is to be docked; and this was done for the *Hood*, both in the case of the original design and again later when the protection and displacement were increased.

The *Hood* was successfully launched in August, 1918, at Clydebank, the ceremony being performed by Lady Hood, widow of Admiral Hood, who lost his life whilst gallantly leading into action the Third Battle Cruiser Squadron at Jutland.

LAUNCHING WEIGHT, 22,000 TONS.

The launching weight was about 22,000 tons. As the other three ships of the class which were commenced had none of them reached the launching stage at the time of the Armistice, it was subsequently decided not to proceed with them, in view of the international conditions, and the *Howe*, *Rodney* and *Anson* were accordingly scrapped.

The general strength of the ship as a girder had to be thoroughly investigated, both the original design and again when the protection and displacement were increased. Assuming the ship among waves equal to her length from crest to crest and one-twentieth of that length in height (a condition which I do not think is ever likely to be met with), the maximum tensile stress on the top of the girder is less than 10 tons per square inch, and I may add that not one of our recent ships has ever exceeded this amount, although a slightly greater stress has been accepted in some recently designed Atlantic liners. The maximum tensile stress on the keel is less than the above, and the maximum compressive stress does not exceed 9 tons per square inch.

GENERAL OUTFIT.

The following are some particulars of the general outfit:

Dynamos.—Eight dynamos are provided and are widely separated, two are driven by Diesel engines, two by turbo engines, and four by reciprocating engines.

Hydraulic pumping engines.—Four hydraulic pumping engines are provided for working the turrets; these and the other more important auxiliary engines are kept well away from the ship's side, as are also the necessary feed tanks.

Pumping and flooding arrangements.—Special consideration has been given to this item in the design of the ship, electrical bilge pumps have been provided, and also some 100-ton submersible pumps to deal with any

water that may enter the ship. For fire and wash deck purposes a fire main is run all fore and aft under protection with rising mains at intervals. This fire main is charged from the 50-ton electrical pumps, and also from the 75-ton steam-driven fire and bilge pumps in the engine and boiler-rooms.

In the boiler-rooms 1,000-ton turbo pumps are fitted; in each engine-room the circulating pumps are arranged to draw from the bilges instead of from the sea, if required. Steam ejectors, each of 300 tons' capacity, are provided in the auxiliary machinery compartments and submerged torpedo rooms.

Steering gear.—The steering gear is of the right and left-hand screw type, with connecting rods attached to a crosshead on the rudder-head. Two steering engines are placed in the after engine-room with shafting led through the shaft passages and thence aft to the steering gear. Auxiliary steering on the Williams-Janney principle worked by electric motor is placed in the after-steering compartment. The main telemotor system is continued aft, so that this gear can be worked from the main steering positions of the ship. The following steering positions are provided: Conning tower, lower conning tower, after engine-room, steering compartment (auxiliary only).

Steering by hand power is not provided, as it has proved to be of little value with these very big, high-speed ships.

Capstan gear.—The anchors, three in number, each weigh 9½ tons, and the chain cable is of 3⅜ inches diameter. Two cable-holders and a middle-line capstan are provided, all connected to the capstan engine. A third cable-holder is provided for the sheet cable for letting go only. At the stern a 3-ton anchor is provided, and the after capstan is worked by Williams-Janney gear driven by an electric motor.

VENTILATION.

Special attention has been paid in our more recent ships, and perhaps more than any in H.M.S. *Hood*, to the system of ventilation, which has been worked out on the latest and most scientific principles; a great deal of experimental and research work having been done to get the best results. The main points to be noted are as follows:

The engine-rooms are ventilated by four 30-inch and three 40-inch supply fans, and four 35-inch and two 50-inch exhaust fans, electrically driven. Each of the auxiliary machinery compartments has two 17½-inch electrically-driven exhaust fans with "natural" supply trunks, all openings being well above the weather decks, in order that ventilation may go on as usual, even in bad weather—a very important point for the comfort and health of the crew. A very complete system of ventilation has been supplied to the seamen's heads, and to all w.c.'s, pantries, sculleries, paint stores, wash places, etc.; the ventilation being generally by means of exhaust fans leading direct to the open air. The living spaces generally are ventilated by trunks supplied by electrically-driven fans. These trunks have openings so arranged that the air is delivered at a low velocity, and strong draughts which might be objected to are thus avoided and a gradual supply of fresh air, keeping the whole place sweet, is thus ensured. Where necessary, the air is passed through steam-heaters for purposes of warming. In all these arrangements care has been taken to maintain as far as possible the transverse watertight subdivision of the ship, by avoiding the piercing of watertight bulkheads. Each of the main transverse compartments is accordingly ventilated independently of any of the other

main compartments, and thus the risk of water finding its way from one compartment to another in case of damage is eliminated.

Accommodation.—In all our recent designs great attention has been paid generally to the accommodation and comfort of the officers and men. Just before the war a special committee was appointed to inquire into the accommodations of ships, and a report was presented, and all recommendations of this committee have, as far as possible, been embodied in our recent ships. The exigencies of the war and the additional requirements which had to be met as the war progressed, to some extent prevented all the recommendations of the committee being carried out; but in the *Hood* practically the whole of these have been embodied and great improvements have thus been made, recreation rooms being provided, and greater facilities for cooking and similar operations. In fact, a modern ship is, in many respects, a home of luxury compared to ships even ten or fifteen years old.

Electrically-driven boat hoists.—Boat hoists, fitted with variable speed control gear of the Williams-Janney type, driven by electric motors, are provided, and a weight of 16 tons, that of the heaviest boat, can be lifted and lowered at a rate of 60 feet per minute.

GENERAL DESIGN.

As regards the general design of the ship, the *Hood* may be cited as an example of what can be achieved by going to a large size. The endeavor has been in her design to embody the armament and armor protection of a first-class battleship including also good under-water protection against torpedoes, and at the same time to give her the speed of the fastest battle cruisers. It has only been possible to do this by going to a great length and displacement. The under-water bulge protection is additional to anything provided in our pre-war dreadnoughts, although these ships had internal bulkheads. The addition of the bulge, which has entirely superseded the provision of torpedo netting, involved a considerably increased weight over and above that of a ship of ordinary form; and although this additional weight was accompanied by a somewhat greater amount of additional buoyancy, it still involved an increased displacement with the accompanying increase in resistance to propulsion.

It may be argued that, in such a large ship, a great many eggs are placed in one basket, and that a very expensive one, but it would have been quite impossible to combine the speed, armament, and protection in a smaller unit.

In connection with the size of the *Hood* and general considerations of design, it is, I think, interesting to note that of recent capital vessels built, and taking the chief characteristics of two classes of about the same displacement, in *Queen Elizabeth* we had a well-armored ship of about 28,000 tons with eight 15-inch guns, and speed of 25 knots. In the *Renown* and *Repulse*, of slightly less displacement, viz., 27,000 tons, though of greater length, we had vessels with 7 knots more speed than the *Queen Elizabeth*, but with only six 15-inch guns against eight, and approximately about half the armor protection provided in the *Queen Elizabeth*. In the *Hood* we are providing the same armament, viz., eight 15-inch guns, as in the *Queen Elizabeth*, armor protection fully equal to, and in fact rather heavier in the aggregate than that of the *Queen Elizabeth*, 6 knots more speed than the *Queen Elizabeth*, which makes the speed nearly equal to that of the *Renown* and the *Repulse*, and, in addition, a full bulge protection against torpedo attack.

AMERICAN BATTLE CRUISERS.

It is of interest to note that the American battle cruisers, which were originally designed with very light armor protection, have recently been modified in the direction of additional protection, making them thus more like the *Hood*, and in the Annual Report of the Chief of the Bureau of Construction and Repair for the fiscal year 1919, he says: "On June 24, 1919, the General Board recommended that battle cruisers be built as a distinct type, but the designs be changed to provide materially greater protection against gunfire and under-water attack, the resulting increase in displacement and reduction of speed, probably less than 2 knots, being accepted." This was approved, and the new plans necessary were got out. It is, I think, satisfactory that the American authorities, who have had full knowledge of what we have been doing in this country in the design and construction of warships, should have thus come to the conclusion that we were on the right lines, and have modified their design accordingly.

COST £6,000,000.

As regards cost, which is about £6,000,000, this, of course, is an enormous sum, but when the displacement and size of the ship and her qualities are considered, I do not think it is very excessive, in view of the present increase in prices. Most things have doubled in cost, and many of our pre-war capital ships, complete with their armaments, etc., cost, on the average, about £90 a ton of displacement. The *Hood* is costing about £145 per ton, or an increase of only 60 per cent above pre-war prices, although she has qualities possessed by no earlier capital ship.

A good deal has been written and talked of lately about the surface capital ship being dead and the necessity for submersibles. But with our present knowledge it would be quite impossible to design a submersible ship which, on the same displacement and cost, had anything like the fighting qualities on the surface which are possessed by the *Hood*. Every ship is a compromise, and if in addition to the ordinary qualities of a battleship she is required to submerge, or even partially submerge, a very considerable percentage of weight has to be added to give her this additional capability of submergence. She becomes still more of a compromise, and the added weight must detract from the fighting qualities of the ship when on the surface, so that, whatever is done, other things being equal, the submersible ship must be inferior to a surface ship in an ordinary action. There are many difficulties of details in the design of a submersible battleship which would take too long to go into fully now, and although there is no doubt that submarines are capable of great development, a little thought will make it clear to anybody that if naval warfare is to continue, the surface ship of the line must still hold the field as the principal fighting unit of any great navy. This view is apparently shared by other countries who are developing their navies, and both Japan and the United States are building large capital surface ships.

I should like to take this opportunity on behalf of my staff and myself of thanking the contractors for the assistance they have given us in getting out working drawings of the *Hood;* this applies not only to John Brown & Co., who have so successfully constructed this, the largest ship ever built for H.M. Navy, but also to the other three firms who, in the earlier stages, helped us with the plans, until the time came when it was decided not to proceed with the three ships ordered from them.

In addition to the printed paper, Sir Eustace d'Eyncourt said:—

"Since my paper was printed, the *Hood* has been undergoing a very exhaustive series of trials which are not, however, yet completed, nor have the results of the trials which have already taken place yet been thoroughly analyzed.

"However, the speed trials at normal draught were completed last week and very excellent results obtained.

"On the full power trial, at a displacement of 42,200 tons, which is considerably in excess of the normal load draught, a speed of 32.07 knots was obtained, as a mean of runs taken on the measured-mile course off the Isle of Arran in deep water. This was obtained with a mean horsepower of somewhat over 150,000 and with 206 mean revolutions of the propellers. The day was not altogether satisfactory, as there was a very high wind—about force 6.

"The engines worked very satisfactorily and a power of between 145,000 to 150,000 was maintained for a period of about seven hours. It is also interesting to note that on the two-fifths power trial a mean speed of over 25 knots was obtained. The propulsive coefficient was in all cases satisfactory, varying from 53 to 55 per cent, and a very satisfactory fuel consumption was also obtained.

"The ship, therefore, with the 5,000 tons increased protection, now has the original speed of 32 knots, which was that laid down for the lighter ship of 36,000 tons. Her full speed, with full fuel and all weights and stores on board, will be at least 31 knots with full power."—"Shipbuilding and Shipping Record."

EXPERIENCE AND PRACTICE IN MECHANICAL REDUCTION GEARS IN WARSHIPS.*

By Engineer-Commander H. B. Tostevin, D.S.O., R.N.

The most important change in the steam machinery of warships of high speed and power during the war period was the departure from the direct-drive turbine installation to that in which the power of the turbines was transmitted to the propeller shaft through mechanical reduction gearing, and it is the object of the present paper to trace briefly this development and indicate the experience that has been gained and also the general practice of the Admiralty as regards the details of these gears. Before the actual adoption of gearing it had long been recognized that some form of reduction device was necessary in order that the turbines and propellers should run at speeds which secure their best efficiencies, and all designs with the direct drive installations had been of the nature of a compromise in which the maximum efficiencies of both turbines and propellers had been sacrificed in order to obtain the best overall results to meet the nature of the particular case. There were three practical methods of obtaining this reduction, viz., hydraulic, electrical, and mechanical, each possessing its peculiar merits, and in the early history it is noteworthy that the first two were more seriously considered than the third, as they did not at the time offer such difficulties in the way of manufacture and reliable transmission of high power. At the present stage in British naval practice the first two methods have not been tried, at least in connection with high power steam machinery.

Whatever method is adopted a loss must occur between the turbines and the propeller shaft from various causes, which need not be gone into

* Paper read before the Institution of Naval Architects, March 26, 1920.

here. Experiments have shown that with the best designs of the respective devices, the losses in transmission may be approximately taken as: hydraulic, 10 per cent; electrical, 8 per cent; mechanical gearing, 1½ per cent. If double reduction gear were fitted the loss with mechanical gear would be about 3 per cent.

The superior efficiency of the mechanical method has been an important factor in reaching a decision as to the type of reduction device to be adopted for naval use, and when it is combined with considerations of simplicity and reliability and the very small cost of upkeep, the combination has been considered up to the present to outweigh the advantages possessed by the other forms of transmission, such as the absence of astern turbines, and the possible maintenance of a higher turbine speed and efficiency at cruising speeds. With the design of turbine as now developed, in which special arrangements are made to permit it to maintain its efficiency to a marked extent as the speed of revolution falls, this latter advantage is to a certain extent discounted.

The endurance and reliability which are features of turbine installations are even more pronounced when geared turbines are fitted, because of the smaller and more rigid structures and moving parts. Also the larger and more efficient propellers improve the maneuvering power, and in particular the stopping effect is increased by the ease with which the larger propellers get a hold on the water. The necessary astern power (which in most warships is approximately one-third of the ahead power) can be obtained on a single velocity-compounded turbine wheel without working the boilers to full power. The smaller turbines and their more substantial construction render them less liable to distortion troubles than is the case with the larger direct-connected units.

To trace the development of mechanical gearing for marine purposes one need go back no further than 1910, when Sir Charles Parsons, who was desirous of applying the steam turbine to vessels of moderate and low speed, carried out his experiments in the *Vespasian*, an old cargo steamer, experiments which have proved to be epoch-making. The success of these experiments, which were dealt with in a paper read before this Institution in 1910, indicated that increased efficiency could also be obtained by means of reduction gearing in ships of the classes from which the reciprocating engine had been definitely displaced by the turbine. The Admiralty made its first step in 1910, and in two torpedo-boat destroyers of that year's program, the *Badger* and *Beaver*, gearing was fitted for part of the main turbine installations. The low-pressure turbines drove the shafts direct, but on a forward extension of their rotor spindles gear wheels were fitted which were driven through pinions by the small fast-running cruising and high-pressure turbines. The maximum horsepower transmitted through each set of gearing was about 3,000. Before these vessels were completed a further advance was made, and in 1912 it was arranged, in two torpedo-boat destroyers, the *Leonidas* and *Lucifer*, of 22,500 horsepower, for the whole power to be transmitted through two sets of gearing. The general arrangement as regards the component parts of an all-geared set adopted in these vessels has been adhered to in naval practice to the present day, viz., each set of gearing is driven by two pinions, driven respectively by the high-pressure and low-pressure turbine. In some installations an additional turbine for cruising has been fitted on an extension of the high-pressure spindle.

On completion of the *Leonidas* in 1914 exhaustive trials were carried out, and the results obtained as regards the efficiency of the machinery installation at all powers were very satisfactory and superior to any that

GENERAL ARRANGEMENTS OF GEARED TURBINES IN BRITISH WARSHIPS.

Fig.1.
BATTLE-CRUISER.
PLAN OF MACHINERY INSTALLATION.

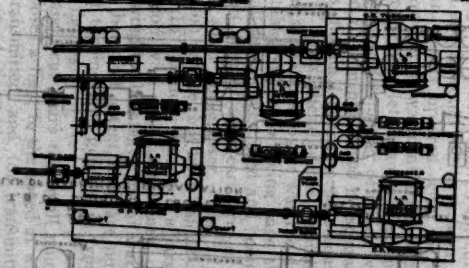


Fig.3.
LIGHT CRUISER (H.M.S. CALLIOPE).
ARRANGEMENT OF TURBINES & GEARING.

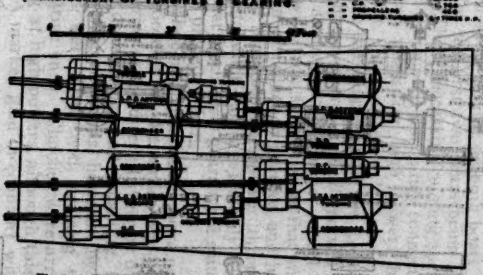


Fig.2.
LIGHT-CRUISER
PLAN OF MACHINERY INSTALLATION

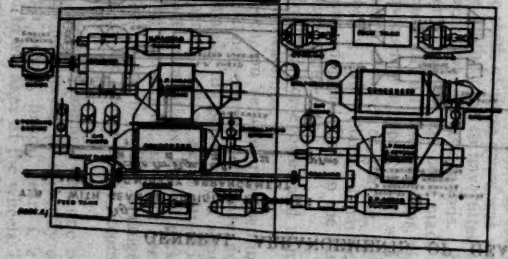
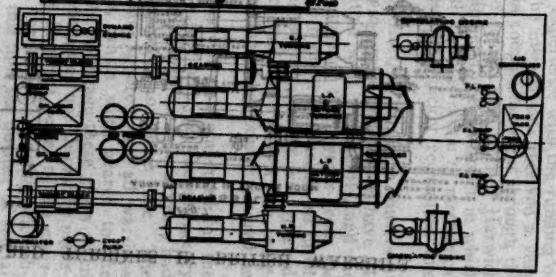


Fig.4.
TORPEDO BOAT DESTROYERS
PLAN OF MACHINERY INSTALLATION



GENERAL ARRANGEMENTS OF GEARED TURBINES IN BRITISH WARSHIPS.

Fig.5.
T.S.S. WITH GEARED CRUISING TURBINES
PLAN OF MACHINERY ARRANGEMENT.

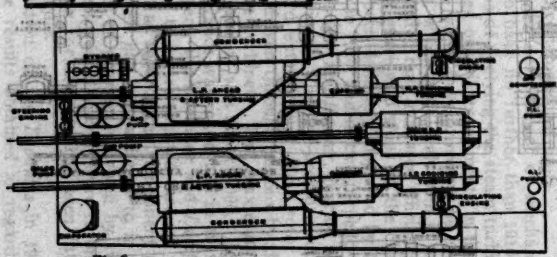


Fig.6.
T.B. Ds LEONIDAS & LUCIFER.
PLAN OF MACHINERY INSTALLATION.

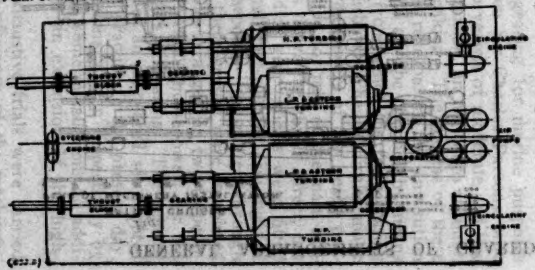


Fig.7.
P.C. VESSELS.
ARRANGEMENT IN ENGINE ROOM.

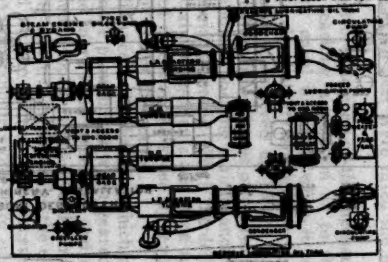
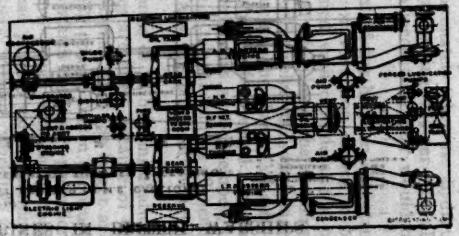


Fig.8.
P. VESSELS.
ARRANGEMENT IN ENGINE ROOM.



had been obtained with direct drive units. Both vessels were hurriedly put into commission in August, 1914, when experience with the running of high-powered gears was practically *nil*. The installations received a very severe test on service, as both boats were in commission right through the war, for the first two years being attached to the hard-worked Harwich force. Their successful running speaks volumes for the manner in which several recognized difficulties connected with the application of mechanical reduction gears for high powers had been so quickly recognized and overcome.

Before the completion of the *Leonidas* and *Lucifer*, two light cruisers of 40,000 horsepower, viz., *Calliope* and *Champion*, were arranged to have all-geared units, but the former vessel was fitted with four shafts, and the latter with two, so it is seen that in the course of three years from the adoption of reduction gearing, and in the third order of warship so fitted, a figure of 20,000 horsepower through one set of gearing was reached. This installation ran satisfactorily on service, and as a result it was recognized that mechanical gears were suitable for the higher power required on a single propeller shaft.

The all-geared installations were not at once universally adopted for the vessels ordered on the earlier war programs, the policy in the emergency of adhering to the direct drive type of turbine, which for several years had proved its efficiency and reliability, being followed for some time. The subject of gearing was, however, by no means ignored, and all warships ordered were fitted with cruising turbines, which were geared to the direct-drive units. Clutches were arranged to disengage these gears when running at high powers, although this procedure was not absolutely necessary. A marked gain in economy within the limits of power of the cruising turbines, and a consequent increase in the radius of action was obtained. It should also be understood that at this time the number of special "hobbing" machines required for cutting the teeth of these gears was very limited, and it would not have been possible at the time to have fitted all the war vessels ordered with the all-geared type of installations. By 1916, however, it was considered that sufficient progress had been made to warrant the complete change over, and in practically all fast warships, viz., battle cruisers (a number of which were subsequently not proceeded with) designed to transmit 36,000 horsepower through each set of gearing, light cruisers, flotilla leaders, torpedo-boat destroyers, "*K*" Class submarines, and "*P*" boats, the all-geared installation was adopted. Only in the case of a few special destroyers was the direct drive adhered to.

The table on the following page shows the total horsepower and number of all-geared sets fitted and being fitted in warships.

Of this number 596 gears are, or have been, on service, representing 6,794,000 shaft horsepower. The 36,000 shaft horsepower sets have not yet been on service, but have run at approximately full power for a short period. In addition there are a large number of sets of gears fitted in conjunction with cruising turbines, or as part installations, but as these gears, owing to the exigencies of war and other conditions, have not been used to their full extent, they are not included in the list. The arrangements of turbines and gearing in typical vessels are shown in the accompanying figures.

In straight-cut gears the small number of teeth in contact, viz., one or two, imposes a limit to the speed at which high powers can be smoothly or safely transmitted, and for transmitting high power, helical gear, *i. e.*, toothed wheels in which the teeth are inclined to the axes of the wheels, is necessary. All the gearing now in use in warships is of the double helical

Type of Vessel.	S.H.P.	Number of Shafts.	S.H.P. per Gearing Unit.	Total Number of Gears.	Total S.H.P.
Battle cruiser ..	144,000	4	36,000	4	144,000
Light cruisers ..	40,000 to 80,000	2 or 4	10,000 to 20,000	90	1,760,000
Flotilla leaders ..	40,000	2	20,000	26	520,000
Torpedo-boat destroyers	27,000 to 30,000	2	12,500 to 15,000	368	5,000,000
"K" Class submarines	10,000	2	5,000	36	180,000
"P" and "P.O." boats.	3,500	2	1,750	128	224,000
Totals ..	—	—	—	652	7,828,000

type, a right and left-hand helix being used to balance the axial thrust entailed. The involute tooth form is invariably used, as it possesses essential advantages over other forms, the most important of which are: (1) The teeth can be generated automatically in a machine by a straight-sided cutting tool, which can be made and ground very accurately. (2) A variation in distance between the centers of pinion and wheel (and such variations occur in marine practice owing to the necessity for ample clearance in the high-speed bearings, and also possibly with misalignment and wear of bearings) does not affect the velocity ratio of the wheel and pinion or the smooth working of the teeth.

In the early days of turbine gearing there was not much experience available bearing on helical gearing for comparatively high powers, and in the first designs the gears were arranged with a helical angle of 20 degrees in conjunction with a pitch normal to the line of the teeth of ¾ inch. This design was rather noisy, and in consequence the angle in succeeding designs was increased to about 45 degrees, which value was known to have proved satisfactory in De Laval installations of small power. The 45-degree design was continued until recently with generally satisfactory results, but in the meantime, with the improvements in gear-cutting processes, it had gradually been established that the noise and tremor experienced in the early installations was more due to irregularities in cutting than to the small helical angle itself. An angle of 30 degrees, which possesses advantages in the way of efficiency and tooth strength, in that the contact between the teeth is increased as compared with 45 degrees, while the normal load on the teeth is decreased, has now been generally adopted by the Admiralty. This angle is rarely an exact number of degrees, as it is affected by the driving and change-wheel mechanism of the hobbing machine, but is as near the desired angle as these details allow. Fig. 16 shows a set of gear wheels and pinions cut to an angle of 32 degrees.

The angle of obliquity used is 14½ degrees, in agreement with the almost universal commercial standard developed for spur gearing with involute teeth. There is no particular virtue in this angle within limits, and the odd ½ degree, it is understood, was decided by the fact that sine 14½ degrees being 0.250, calculations were simplified.

Fine-pitched gears give more silent running at high speeds, and a normal pitch of 7/12 inches has been adopted for all but the very largest installa-

FIG. 15.

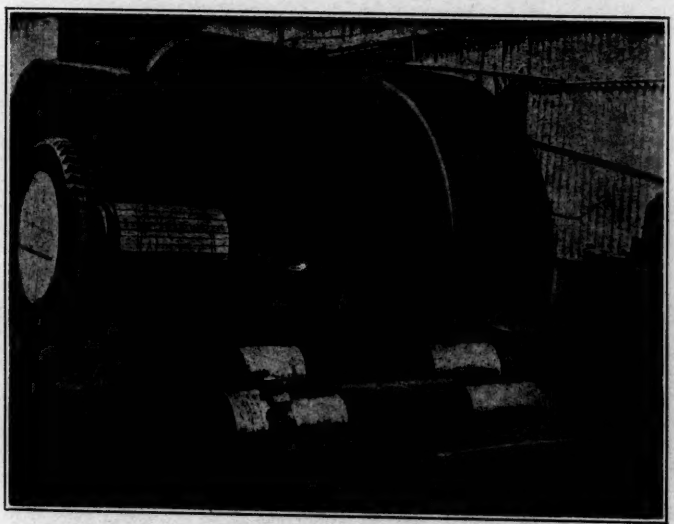
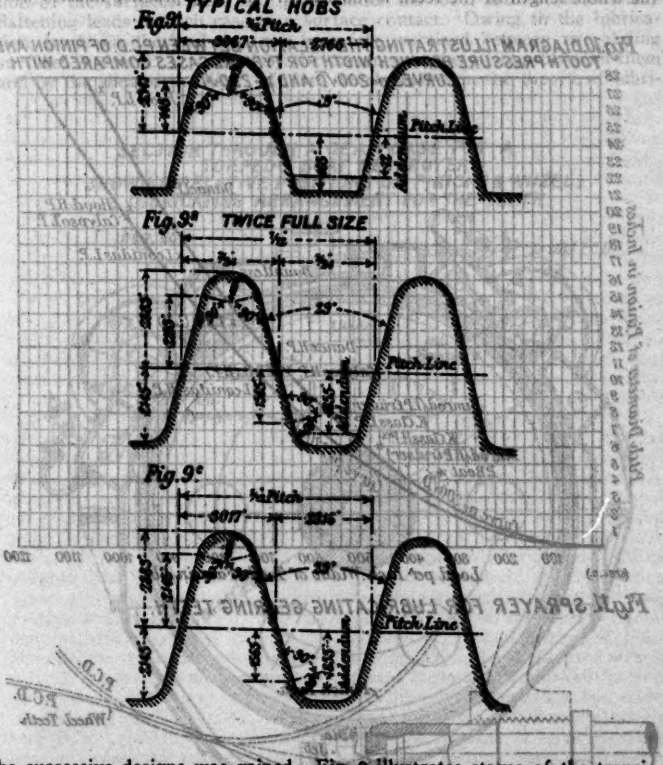


FIG. 16.

tions. While the pitch and obliquity have remained the same, the proportion between addendum, dedendum, and pitch, respectively, and the shape of the root and tip have been changed from time to time as experience of



the successive designs was gained. Fig. 9 illustrates stages of the transition, it being noted that the form of cutting tool is shown, and that while the height and normal pitch of the actual tooth generated will be the same as for the hob, the thickness and contour of the teeth will depend upon the pitch circle diameter and upon the hob design. The earliest design of hob (which is not shown) produced a form which gave at the base of tooth only a small radius, and it was at a very early stage modified to provide a more pronounced radius at the tooth root. In the design in Figs. 9b and 9c the rounding of the tips of the teeth was introduced to overcome the "digging in," and excessive wear of the tips which had been observed in some designs, probably due to the failure of the oil film on the surface at this position.

The question of the lubrication of the teeth is very important, and while running, an ample amount of oil, which is generally injected from nozzles on to the line of contact of the teeth, is necessary. It is usual to arrange ¼-inch to ⅜-inch nozzles of about 5-inch pitch, discharging the oil under a pressure of from 5 pounds to 10 pounds, the jet being fan-shaped, so that the whole length of the teeth is lubricated. In early practice nozzles were

Fig. 10. DIAGRAM ILLUSTRATING THE RELATION BETWEEN P.C.D. OF PINION AND TOOTH PRESSURE PER INCH WIDTH FOR TYPICAL CASES COMPARED WITH CURVES $p = 200\sqrt{D}$ AND $p = 220\sqrt{D}$.

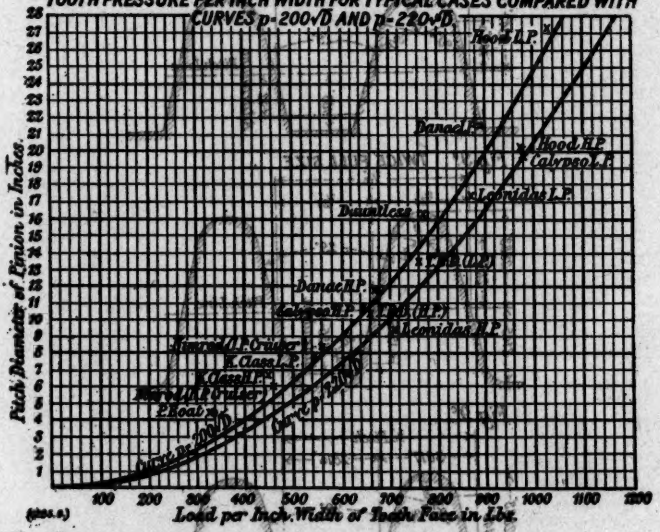
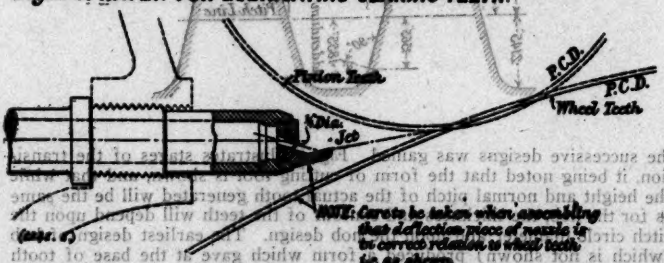


Fig. 11. SPRAYER FOR LUBRICATING GEARING TEETH.


Pinion Teeth.
P.C.D.
P.C.D.
Wheel Teeth.
Dia.
Jet
NOTE: Care to be taken when assembling that deflection piece of nozzle is in correct relation to wheel teeth, i.e. as shown.

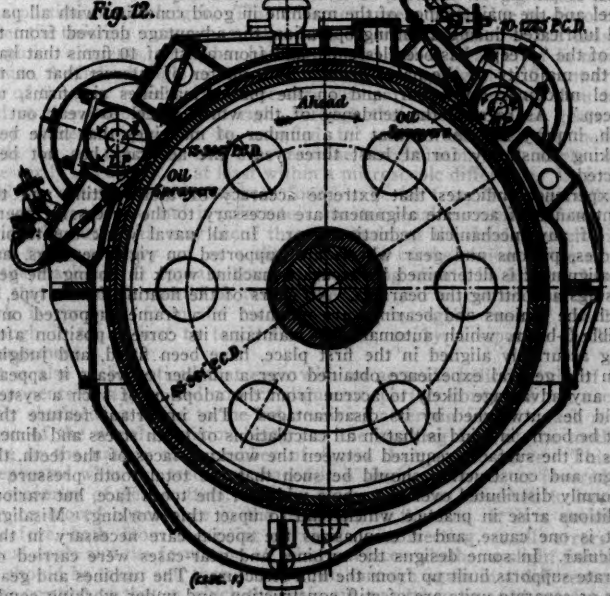
fitted to discharge to the line of contact both on the entering side when running ahead and also when running astern, but the latter groups are not now fitted, as experience shows they are not required, considering the short periods of astern working usual in naval vessels.

Figs. 11 and 12 show a type of sprayer fitted, together with its arrangement on the gear-case.

The teeth of gearing engage by line contact, but it must be appreciated that contact under a load can no more occur on a line than the load on a ball bearing can occur on a number of geometrical points. Elastic deformation of the surfaces must take place under the load, and the momentary flattening leads in each case to a surface contact. Owing to the lubrication, however, the load that can be safely sustained between two sliding surfaces in contact is decided by the pressure in the oil film between them and by the area of the film. Modern developments in the theory of lubri-

SECTION THROUGH A GEAR CASE FOR A TORPEDO BOAT DESTROYER SHOWING RELATIVE POSITIONS OF PINIONS & WHEEL, & LUBRICATING ARRANGEMENT FOR THE TEETH.

Fig. 12.



cation seem to show that the pressure in an oil film between two surfaces varies as the thickness of the film, and that beyond a certain thickness of film no pressure is maintained. For a given thickness of film it will extend over an area practically proportional to the square root of the radius of curvature of the surface, and this area may be regarded as the virtual area of tooth contact. The radius of curvature may be taken on the pinion teeth alone, as the gear-wheel teeth are sensibly flat, and as this average

radius of curvature of the tooth form varies as the pitch diameter of the pinion, an empirical rule depending to a certain extent on the helical angle for the allowable tooth pressure $p = K \sqrt{D}$, where K is constant, is obtained. p being the intensity of pressure in pounds per inch of axial length. The values of K for a large number of cases are shown in Table I, and are seen to be round about 200 to 225 in the majority of cases. The pressures obtained per inch width of face are also shown, and vary from 700 to 1,080 in the high-powered installations at the nominal full power. Table I and Fig. 10 illustrate these points in typical cases.

A difficulty in the early history of gear-cutting was caused by the fact that the master worm wheel which rotated the gear wheel during the cutting operation was not quite accurate; this difficulty led to an addition to the machines of a "creep" mechanism, by means of which the job is rotated at a higher speed than the table. Any recurring error in the worm wheel which might be copied on the job being cut is, with such a machine, no longer in the direction of the axis, but is distributed in helices on the gear wheel or pinion. With improvements in the accuracy of this master worm wheel and the maintenance of the machine in good condition with all parts well lubricated during a cutting operation, the advantage derived from the use of the "creep" has been lessened, and from a list of 10 firms that have cut the majority of the naval gears it is a matter of interest that on the wheel machines only four, and on the pinion machines six firms, use "creep." As regards the tendency of the worm wheel to wear out of truth, inquiry has shown that in a number of machines that have been working constantly for at least three years the accuracy has not been affected.

Experience indicates that extreme accuracy of tooth-cutting and the maintenance of accurate alignment are necessary to the successful operation of any mechanical reduction gear. In all naval work the turbine spindles, pinions and gear wheels are supported on rigid bearings, and the alignment is determined by accurate machine work in boring the gear housings and fitting the bearings. No gears of the floating frame type, in which the pinions and bearings are mounted in a frame supported on a flexible I-beam, which automatically maintains its correct position after being accurately aligned in the first place, have been fitted, and judging from the general experience obtained over a number of years it appears that any advantage likely to accrue from the adoption of such a system would be outweighed by its disadvantages. The important feature that must be borne in mind is that in all calculations of tooth stress and dimensions of the surfaces required between the working faces of the teeth, the design and construction should be such that the total tooth pressure is uniformly distributed over the whole width of the tooth face, but various conditions arise in practice which tend to upset this working. Misalignment is one cause, and it emphasizes the special care necessary in this particular. In some designs the turbines and gear-cases were carried on separate supports built up from the hull structure. The turbines and gear-cases as separate units are of stiff construction, and under working conditions do not suffer distortion. The possibility of relative motion between the turbines and gear-case existed, and although nearly all vessels so constructed experienced no troubles with the gearing, yet in two torpedo-boat destroyers the gear became very noisy, and wear took place on the teeth. Although no actual breakdown occurred, the defects were objectionable, and the gears were removed for dressing up. When the alignment was again tested it was conclusively proved that relative displacement of the gear-case and turbines had actually taken place. In later designs this

movement is not possible, as the after-ends of the turbines are now supported on stiff extensions of the gear-case, the whole installation forming a rigid unit. This is illustrated in Fig. 15.

Owing to the action between the teeth when transmitting power, the pinion and wheel tend to increase their distance between centers. This, as pointed out, does not affect the smooth or regular action of involute teeth, but the line of centers through turbine and pinion should be free to remain parallel with the axis of the wheel, and the clearances in all the bearings should permit of this, otherwise misalignment will take place. Another cause that arises is that due to the deflection of the pinions and wheel, which in each case is of a double nature, viz., that due to the twist and that due to the bending. These points should not be overlooked in any design, but with the proportions brought about by other considerations, these deflections are not such as materially to affect the working of a correctly constructed gear. Apart from the above causes it has been observed, in many gears that the load has not been distributed over the whole length of the teeth, an undesirable concentration of pressure having taken place on a portion of the length. This does not appear to be due to distortion under the twisting moment, as it has been observed at either of the ends of the pinion elements, and it has been attributed rather to an error in manufacture, the helical angle of the pinion element being slightly different from that of the gear wheel. It is necessary that particular attention should be paid to this important point right through the construction of gears, as many makers appear to be satisfied to assume that as this helical angle is a function of the machine mechanism, the angle will necessarily be correct, or at least within a microscopic difference. Unfortunately, in some cases, this has been found to be far from true, especially in the case of pinions, and it points to some torsional movement of the job during the cut. This may occur through strain of the driving mechanism when taking a heavy cut, or a slight slow slip of the pinion in the chuck of the machine. It appears better that nothing should be left to chance, and that the angle should be tested after the gear is cut and before the work is removed from the machine, either by again running the hob through it so as just to touch the surface, or by a separate checking arrangement.

It is also to be noticed that even with an accurate helical angle of the components of a gear and correct alignment, perfect contact is not necessarily obtained, as owing to the hob receiving a certain feed for each revolution of the wheel or pinion being cut, the working face of the tooth is more in the nature of a number of facets than a fair curve. This is accentuated when the teeth are shaped in a single operation, as favored by the majority of gear-cutters, and the faces are often observed to have a torn or scored appearance in the direction of the cut. It is a question, therefore, both from the point of view of correcting small errors in helical angle and the formation of fairer surfaces whether a slight finishing cut should be taken and then the hob run through again in a different relative position to that in which the main cut was taken so as to remove the high points between the facets and to smooth the surfaces that have been torn, due to the hardness of the material when taking the primary cut.

The speed of the teeth has an effect on the running and durability of the gear, owing to the intensity of shock on them. With the modern methods of cutting and manufacture, and with forged materials, the limits formerly used with spur gearing have been much exceeded, and there is ample experience to prove that satisfactory working can be obtained with speeds of 140 foot-seconds, and there seems no reason to assume that this

speed could not be sensibly exceeded. The effect of the rise of temperature on the teeth, due to the rubbing contact when off the pitch line, has not appeared to be sufficient to affect the working in a well-lubricated gear. In general, gearing ratio of 8 or 9 to 1 is not exceeded in naval practice of moderate and high power.

Typical gear ratios from naval practice are given in the table below.

Type.	S.H.P. per Pinion.		Ratio of Turbine Revolutions to Propeller Revolutions.	
	H.P.	L.P.	H.P.	L.P.
Battle cruiser ...	17,500	18,500	7·1	5·2
Light cruiser ...	8,500	11,500	9·2	5·1
Flotilla leader ...	9,000	11,000	5·5	5·0
Torpedo-boat destroyer ...	6,250	7,850	8·5	6·5
"K" boat ...	2,250	2,250	8·7	7·7
"P" boat ...	875	875	11·6	11·6

In the design of a reduction gear the question of relative axial movement between the pinion and the gear wheel is of importance. This movement takes place due to the necessity of the pinion requiring to adapt itself so that the load on each element should be the same. This movement is not difficult to cater for in a single reduction, as the gear wheel being fixed by the thrust block, it is only necessary to provide an expansion coupling between pinion and turbine spindle, and allow clearance between the ends of the pinion elements and the adjacent bearings. The problem is, however, more difficult in the case of certain designs of double-reduction gear, as relative movement is required between the shaft gear wheel and the intermediate reduction element, and also between the latter and the pinion. The intermediate reduction wheel being a heavy fitting, for it carries four working faces, two of fairly large diameter and two small, possesses considerably more inertia than a pinion, and in the event of fitting such a gear in warships this aspect of the problem would require careful consideration.

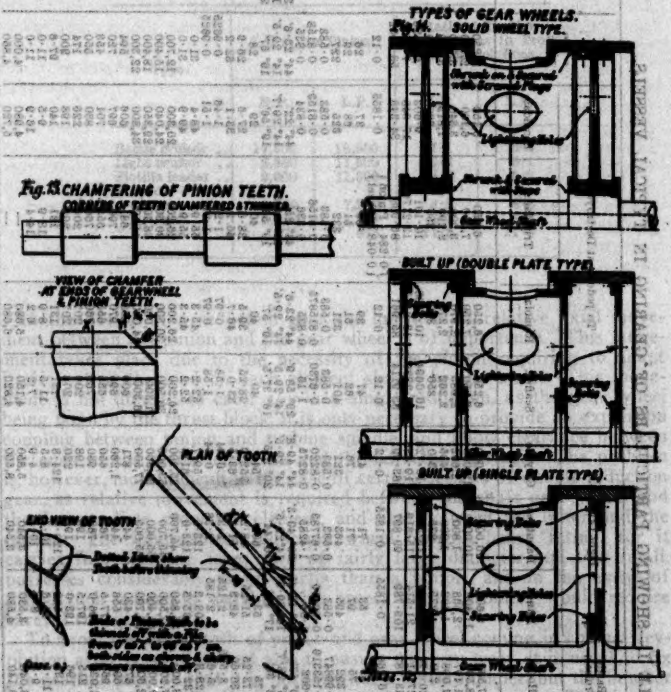
To avoid the same teeth of pinion and gear wheel coming in contact revolution after revolution, and also to ensure uniform wear, it is usual to fix the gearing ratio so that it is not only not a whole number, but the number of teeth in the wheel and pinion have no factor in common. This is seen in Table I.

In addition to the two cases of trouble with gearing previously referred to, there have been a few cases of failure due to the fracture of pinion teeth at their ends. This is the weakest part of the teeth, as no support is given on one side of the portion receiving the pressure. Chamfering off of the ends of the pinion teeth, as shown in Fig. 13, prevents the load coming on the extremity. Fracture of these teeth might be expected with an error in the helical angle of the components, leading to an excessive intensity of pressure at one end of an element, but cases have occurred where a fracture has taken place, and yet an examination of the gears has shown the pressure to be uniformly distributed along the length of the tooth. Seeing that under these conditions the tooth stress is low, other

TABLE I.—SHOWING PARTICULARS OF GEARING IN TYPICAL VESSELS.

		Battle Cruiser.	Light Cruisers.			Torpedo-boat Destroyers.				"E" Boat.	"P" Boat.	Flotilla Leader (Geared Cruiser).	
		Hood.	Dublin.	Dauntless.	Calypso	Seafire.	Rameis.	Trinidad.	Lochinvar.			Nimrod.	
												H.P.C.	I.P.C.
Horse-power	H.P.	17,500	5,000	10,000	5,000	6,250	6,250	6,250	5,150	2,250	675	1,250	5,500
	I.P.	16,500	11,500	10,000	13,000	6,250	7,250	6,100	2,750	675	—	—	—
Revolutions per minute	H.P.	1,427	2,568	1,800	2,562	2,925	2,970	3,000	2,940	2,250	3,500	2,550	2,550
	I.P.	1,090	1,453	1,800	1,435	2,262	2,270	2,290	1,515	2,000	3,500	—	—
	Propeller shaft	216	275	275	240	350	350	350	280	200	300	342	483
Pitch circle diameter, in inches	H.P. pinion	20·174	11·786	13·219	10·723	10·0994	10·1252	10·101	9·605	6·7529	4·82	6·016	6·111
	I.P. pinion	27·91	21·514	15·219	19·238	13·2252	13·2417	13·2176	17·240	7·1218	4·82	—	—
	Gear-wheel	143·787	109·789	99·687	88·861	85·0114	85·061	85·063	84·548	56·8842	56·362	61·804	48·405
Addendum of teeth, inches		0·266	0·1855	0·1855	0·12	0·12	0·12	(0·234 Pinion / 0·048 Wheel)	0·1855	0·12	0·168	0·1855	0·1855
Number of teeth	H.P. pinion	85	82	71	41	47	39	39	37	26	24	28	21
	I.P. pinion	75	97	71	76	48	51	51	66	29	24	—	—
	Gear-wheel	382	433	403	331	401	331	403	325	227	221	149	185
Pitch, in inches	Normal	0·09647	0·0602	0·0602	0·0602	0·0585	0·0583	0·0583	0·0583	0·0583	0·04196	0·0585	0·0583
	Circular	1·155319	0·09679	0·07722	0·0230	0·0780	0·015675	0·0156	0·0156	0·0156	0·009	0·0225	0·0225
	Axial	2·002	1·04	1·4285	0·0275	1·16	0·085	0·536	0·534	0·335	1·04	0·0225	0·0225
Helical angle		20° 57′	20° 12·4′	20° 40·5′	44° 49·5′	29° 58·5′	44° 23·5′	44° 23′	44° 22′	44° 23·5′	20°	44° 49·5′	44° 49·5′
Tooth angle	Normal	14° 30·7′	14° 30·5′	14° 23·5′	16° 29·5′	14° 29·5′	14° 29·5′	14° 29·5′	14° 29·7′	14° 29·5′	17° 30′	14° 29·5′	14° 29·5′
	Circular	14° 30′	17° 12′	16° 44′	20° 1·5′	16° 37·5′	16° 37′	16° 40′	16° 40′	16° 37′	20° 0′	20° 0′	20° 0′
Width of pinion face, in inches	Total	75	53	60·5	54	40	40	40	30	28	21	24	35
	Effective	73·25	51·25	58·25	52·25	38·25	39·5	38·6	27·6	26·6	20·5	24	35
Number of teeth engaging		36·63	48·3	61·7	68·5	51·0	46·7	46·2	33·1	33·2	19·71	29·2	42·6
Length of teeth contact, in inches, per tooth	H.P.	2·615	2·125	2·38	0·94	1·58	0·97	1·51	1·46	0·9625	1·79	1·527	1·98
	I.P.	2·63	2·155	2·38	0·94	1·58	0·97	1·51	1·51	0·9625	1·79	—	—
Total length of contact, in inches	H.P.	189·8	103·8	133·6	50·7	82·2	45·3	82·9	48·4	31·0	35·80	39·08	59·2
	I.P.	192·9	105·5	133·6	51·4	82·2	45·3	55·9	49·9	31·0	35·80	—	—
Tangential load, in lb.	H.P.	79,000	39,650	45,700	25,200	23,200	24,200	24,200	20,200	12,700	6,342	11,000	19,500
	I.P.	77,800	47,500	45,700	45,000	20,500	20,500	20,500	24,040	15,400	6,342	—	—
Normal load, in lb.	H.P.	87,000	44,400	55,000	31,900	21,500	26,000	26,100	22,350	13,400	6,803	10,150	24,000
	I.P.	82,600	52,700	55,000	57,500	26,500	44,000	44,100	24,800	22,300	6,803	—	—
Load in lb. per inch, inch length of contact	H.P.	575	430	445	571	600	898	661	606	594	224	413	482
	I.P.	571	450	445	1,260	645	970	795	497	720	224	—	—
Load, per inch, on total width of tooth face in in. (= P)	H.P.	1,160	777·5	758	960	755	1,455	1,071	880	850	338	756	857
	I.P.	1,080	884·0	758	960	760	760	760	880	850	338	—	—
K in formula P = K √F.D.N.	H.P.	215	197·5	198	198	200	207	204	236	176	155	187	198
	I.P.	198	198·0	198	220	206	207	205	198	200	155	—	—
Speed of teeth on pitch line in f.s.		125	120	120·4	124·8	721	731	731	745	97·6	70·0	62·5	101
Maximum relative velocity of teeth in f.s.	H.P.	11·8	10·6	10·6	6·4	11·5	9·9	12·0	9·5	11·0	15·3	15·1	14·6
	I.P.	9·5	8·78	10·6	4·9	7·2	9·2	14·4	10·9	11·0	15·3	—	—
Stress at pitch, in lb. per sq. in.	H.P.	3,040	3,630	3,540	3,560	4,150	5,260	—	4,260	4,550	1,960	3,600	4,740
	I.P.	3,140	4,560	3,540	3,560	4,250	5,260	—	5,750	4,550	1,960	—	—

reasons have been sought for these failures. Of the six cases, four occurred during the contractors' trials, one occurred after two and a half months, and the other after seven months' service. In neither of the latter two cases has the defective gear been removed from the vessel, and they are still running with the broken teeth having been trimmed off and the gears dressed up. Defective material has been indicated as the probable cause of this defect, but although careful analysis has been made, in no


TYPES OF GEAR WHEELS.
Fig.14. SOLID WHEEL TYPE.
BUILT UP (DOUBLE PLATE TYPE).
BUILT UP (SINGLE PLATE TYPE).
Fig.15. CHAMFERING OF PINION TEETH.
CORNERS OF TEETH CHAMFERED & TRIMMED.
VIEW OF CHAMFER AT ENDS OF GEARWHEEL & PINION TEETH.
PLAN OF TOOTH
END VIEW OF TOOTH

cases have the results indicated anything seriously wrong with the metal, although it has been noted in one or two cases that the structure has not been the best possible. It should be noted that before the vessel has been accepted by the Admiralty the teeth of the high-pressure pinion in a destroyer have received their full load about 1,000,000 times, and in the case of a light cruiser about 600,000 times, and the causes of failure referred to may, in the absence of any other reasonable cause, be attributed to a fatigue at a flaw developed in the alloy during its manufacture.

The type of steel used for pinions must possess several essential qualities, viz., (*a*) strength to resist permanent deformation under stress; (*b*) stiffness, to avoid undue whip or elastic deformation when running; (*c*)

surface hardness to prevent wear; (*d*) non-brittleness, so that it can stand shocks and blows. In addition, there is the important fact that the teeth have to be cut after tempering the material, as otherwise distortion of the teeth would arise. The specifications governing the materials for gear forgings for naval service and their treatment are given in the appendix.

It appears that finality in respect to the most suitable material for pinions has not been reached, and it is possible in future that the use of a chrome-nickel steel with a low percentage of chromium, say, from 0.3 per cent to 0.5 per cent, will receive consideration for these details in place of the nickel steel now employed. It is known that the effect of very slight variations in the percentage of the constituents, variations that may occur in different parts of the forging under the best conditions, are sufficient to completely upset the anticipated results of the heat treatment, but judging generally from experience with service gears there does not seem any pressing reason to depart from present practice.

Before a vessel is accepted by the Admiralty the gears are opened up after the full power trials, and all surfaces, especially the roots and ends of the teeth, scrupulously examined with a powerful magnifying glass to ascertain whether there are any flaws or signs of cracking present that may lead to subsequent fatigue failures.

During the early running of gears on service there is often observed on the faces of the teeth an action known as pitting. Small flakes of metal are detached from the otherwise smooth surface of the teeth, leaving small hollow depressions, varying in size from a small speck to as large as ⅛ inch diameter, and 15 thousandths to 30 thousandths of an inch deep. They are most common in the region of the pitch line, where only rolling action occurs. It is difficult to account entirely for this pitting. A lack of homogeneity in the material at the tooth surface (and careful analysis by micro-photography does reveal this), or crushing of minute high spots causing failure of small sections of the metal at these high spots are possible reasons for this pitting. This action in the vicinity of the pitch line may possibly be due to the metallic contact recurring there as a result of the failure of the oil film, observing that the rubbing velocity of the teeth is gradually reduced from a maximum at the tips and root to zero at the pitch line. In any case the action is not found to be deleterious to the smooth and efficient working, and it usually ceases after a short time. Deformation leading to scoring and wear of the teeth is a serious, but fortunately rare, fault in naval gears, and the only cases on service in which such wear has taken place were those previously referred to as having suffered from misalignment and in the case of a "P" boat referred to later. It has, however, taken place during the contractors' trials of at least two ships, involving removal with either re-cutting to a modified form of tooth, necessitated by the wear, or the substitution of another gear. The working is seriously prejudiced by such wear, as, apart from the reduction of life involved, it is accompanied by excessive noise and heavy "shivering" of the gear, which transmits its effects to the turbines and subsidiary fittings, causing failure of lubricating pipes, etc. Once deformation is present, or sets in, the whole face of the tooth is affected, even at the pitch line, where the contact is more of a rolling nature, as, due to the wear of the faces of the teeth, there is a tendency for the points of the harder pinion teeth to "dig in" and tear across the flank of the tooth when disengaging, the concentration of the load on the pitch-line then being so great as to cause elastic deformation and a continuation of the digging in, failure of the oil film takes place, and the whole surface of the tooth is scraped away. In the two cases referred to, during trials some

slight error in the tooth form was considered to have been the cause in initiating the action, being then accentuated by the consequent failure of the oil film.

From this same point of view any grinding in of the gears, apart from the practical difficulties of carrying it out, appears to be undesirable on theoretical grounds, and as it causes wear of the flanks and faces of the teeth, it leads to an undue concentration of pressure on the pitch-line. A few service gears have been subjected to this operation, but it is not now being applied.

Another usual feature in gears during their early life is for a minor action of the above nature to take place, the teeth at the regions of higher pressure being rubbed away, or caused to flow so that a sharp, razor-like edge or small "rag" is formed on the points of the teeth. It can be stated as a general principle, however, that once these harder regions of pressure have been "beaten down," the load is more uniformly distributed, and the action does not continue, nor is the working of the gear affected.

With the oils used for naval service corrosion of the gears does not take place. Some slight admixture of water with the oil always takes place, chiefly due to steam from the turbine glands entering the adjacent bearings, but beyond a rusting effect of the wheels and pinions generally the working faces are not affected. It has been suggested that in the early life of a gear it might be advisable to use an oil carrying a proportion of graphite, from the point of view that under the working pressure the microscopic irregularities in the surfaces would be filled by small particles, and in a short time highly polished surfaces obtained. As the lubricating system of gearing is, however, common with that of the turbine installation, it has not been generally adopted, but recently this is being tried in two torpedo-boat destroyers.

From the general experience so far gained there is no reason to suppose that gearing should not have a long life, provided that a reasonable amount of care be exercised in running an installation that has successfully passed through its specific trials. Gauges are supplied for testing the wear of the teeth, but as in practically all cases the wear is inconsiderable, it is not usual to keep records of this.

Some cases of failure having been mentioned, it is of interest to sum up as a whole and see how far the anticipation of reliability has been substantiated. Of the 596 sets of all-geared installations on service in the navy, some extending up to nearly six years, it has only been necessary to actually remove three for refit in the misalignment cases referred to, and here it must be emphasized that no actual breakdown occurred, and the gears, after dressing up and in one case new pinions being supplied, were subsequently re-utilized. Also in the case of one of the "P" boats the two gears had to be removed on service owing to their becoming noisy and wearing slightly. The wheels were "dressed up" and new pinions fitted. Of the cases of fractured teeth, which are two in number, the gears were not removed, the broken or cracked portions being removed and damaged teeth smoothed up. In another case a breakdown occurred through a ⅞-inch bolt, inadvertently left in the gear-case, passing between pinion and wheel and damaging a portion of the length. This set was replaced by a spare, but the old wheel after cleaning up is retained as a spare with 3 inches of the tooth face out of 40 inches total missing. The failures on trials have been four with fractured teeth and two with excessive wear, and it must be admitted that these figures are most satisfactory when it is remembered that the development and experience of these gears for high powers was, except for a few cases, confined to warships.

Apart from the reduction in size of the turbines which led to more rigid and reliable units, there is a great decrease in the quantity of turbine blading required, and at one period of the war, owing to the enormous demands for copper in the munitions, etc., service generally, any question which lessened the demand for the supply of this metal or its alloys was of importance, and the reduction in blading required owing to the adoption of the all-geared installations reduced the strain on the blading makers, and avoided any possible delay in the construction of warships from this cause.

Some typical figures showing this are: In a direct-driven torpedo-boat destroyer of 25,000 shaft horsepower, the total length of blading sections fitted is approximately 70,000 feet, as compared with 7,720 feet in an all-geared installation of 27,000 shaft horsepower. As, however, the section of the blading in the all-geared type is heavier, the weight is not in the same proportion, being as 4 to 8½ tons. Apart from the saving of fuel, and slight reduction in boilers owing to the gain in economy, there is no great saving of overall weight in the all-geared installation, the gearing being a heavy fitting and helping to outweigh any saving obtained on turbines and boilers. Some typical weights are:—

—	—	S.H.P.	Total Weight.	S.H.P. per Ton.
			tons.	
Torpedo-boat destroyer	Direct driven	25,000	372	67·5
	All-geared ..	27,000	393	69·0
Flotilla leader	Direct driven	36,000	554	65·0
	All-geared .	40,000	615	65·0

The question of economy which was one of the primary reasons for fitting reduction gearings has been substantiated. Generally speaking, the advantages of the present all-geared installations over the most recent direct-drive types fitted are shown in the table below, the figures being based on the increase of the radius of action for the same quantity of fuel under service conditions, and therefore taking into account the gain in economy in the turbines and the effect of an increase in propulsive efficiency.

PERCENTAGE IN EFFICIENCY OF GEARED INSTALLATIONS.

—	One-fifth Power.	Two-fifths to Three-fifths Power.	Full Power.
Light cruisers ..	16·5	16·3	16·1
Flotilla leaders ..	20·0	20·0	17·0 at 30 knots
Torpedo-boat destroyers	20·0	20·0	20·0 at 30 knots

The initial steam conditions in the cases of the all-geared and direct-drive installations are the same in each case.

It should also be noted in this particular that super-heated steam has not up to the present been used in any all-geared turbine installations in the naval service, but with the advantages to be gained from the possible adoption in the future the smaller rigid turbines used would conceivably be more adapted for superheat than the larger direct-drive units.

The author is indebted to Messrs. Thornycroft for the photographs of the gearing of a torpedo-boat destroyer and a flotilla leader, illustrating this paper.

APPENDIX.

I. Extract from Admiralty Specifications.

Gearing for Turbines.—The bearings, shafting, and all parts of the gearing in connection with the turbines are to be accurately in line and the pinion shaft is to be truly parallel with the shaft of the main wheel, and the gearing is to be most accurately adjusted for correct working. The whole of the above parts will be tested both in the shop and when erected on board to ensure that these conditions are observed. The supports of the gearing are to be rigid, and the whole system is to be worked without vibration on trials.

When the pinion bearings are bored and fitted in place to give correct oil clearances a circle concentric with the bore is to be clearly marked on the ends of the brasses for guidance in boring after subsequent remetalling.

The gear wheels are to be of forged steel and the rims upon which the teeth are cut are to be separate from the body of the wheels.

Special consideration is to be given in the design and method of construction of the pinion and gear wheels to ensure that the process of manufacture of the parts upon which the teeth are cut is such as to ensure

	Ultimate Tensile Strength.		Elongation in B.S. Test-Piece "C," "D," or "E," as Approved. Not less than, per Cent.	Bending Test. Angle not less than
	Not more than, Tons per Sq. In.	Not less than, Tons per Sq. In.		deg.
Rims of gear wheels	35	31	26	180
Gear-wheel shafts ..	38	34	23	180
Main gearing pinions :				
In the direction of axis of pinion ..	—	40	23	180
Across the axis of pinion	45	40	16 (on C test-piece)	180

maximum uniformity of material and as equal a degree of hardness as possible over the whole of the working surface of the teeth. This condition is to be demonstrated if required to the satisfaction of the Admiralty Inspecting Officers.

The teeth of the gear wheels and pinions are to be chamfered at the ends in accordance with latest practice.

The gearing is to be cut by approved makers and in machines of approved type, and the accuracy of these machines is to be demonstrated, if required, to the satisfaction of Admiralty Inspecting Officers. The lubricating arrangements for the gearing and pinion shaft bearings are to be carefully considered and specially submitted for Admiralty approval.

The teeth of the gear wheels and pinions are to be cut with a spiral angle of 30 degrees at the works of the same maker.

The flexible couplings are to be made independent of rotor spindles and pinion shafts and fitted with provision for lubrication on an approved principle.

The pinion element of the gearing is to be of nickel steel containing about, but not less than, 3.5 per cent nickel and 0.3 per cent to 0.35 per cent of carbon.

The pinion gearing may be formed in separate rims shrunk on to the shaft and suitably secured in an approved manner, or manufactured solid with the pinion shafts. If made in separate rims, the blank is to be hollow forged or rolled, leaving as little margin as practicable for turning off on the internal or external diameter, but at least 1 inch is to be machined off each end of the blank after forging.

If made solid with the shaft, the pinion blank is to be reduced at least 2 inches in diameter and 2 inches on each side of each pinion blank by machining from the finished forging. The pinion shaft is to be hollow, and after rough machining it is to be oil-hardened by the following or other approved process:—

The forging is to be heated to 850 degrees C. and then quenched in oil, this to be followed by tempering at 600 degrees C.:—

In both treatments the time of heating is to be the minimum required to raise the center of the forging to the specified temperature, and this minimum should not be exceeded by more than half an hour.

The gearing teeth are to be accurately shaped, are to be well rounded off at the roots, and are to have approved clearances in the pitch circles.

Test Pieces.—Test pieces are to be taken from each forging, and may be cut off after annealing.

Tensile Tests.—The test pieces are to satisfy the conditions of the preceding table.

II. Gear Wheels.

The gear wheel is made up of a continuous shaft, of the rim, and of a center uniting the shaft to the rim. The center may be made of a solid steel casting, or of two or more forged steel wheels or be constructed of plates.

Fig. 14 shows sections through typical wheels. Cast and forged steel wheels are shrunk and pinned to the shaft, and the rim is in turn shrunk and pinned on to them. Plate centers are bolted by fitted bolts to flanges, made solid with the shaft and with the wheel rim, respectively. The larger designs are arranged with a separate casting keyed to the shaft which carries the flanges of the center plates. The rim is made of forged carbon steel of tensile strength, 31 tons to 35 tons per square inch. The two portions of the rim in which the teeth are cut are made in one piece for those designs in which the rims are shrunk on to the wheel centers comprised of steel forgings or castings. In the built-up designs of wheel centers comprising plates, each rim section and the intermediate portion of the rim joining them is made separately and bolted together. This is necessary for constructional reasons except in very small designs.

No special advantages as regards working on service is claimed for either form of construction, and no cases of failures of wheels or gear cases structurally have occurred on trials or service.

Fig. 13 shows the method of chamfering off the corners of the teeth of the pinions and gear wheels to prevent any possible excessive stress being taken on an overhung or incompletely supported extremity. As a further safeguard to the teeth of the pinions having in view the effect of deflection, they are also thinned off as shown. This thinning off also keeps the load off the end of the wheel teeth.—"Engineering," April 9, 1920.

THE PRESENT TREND OF MARINE ENGINEERING.

By Lieutenant-Commander H. G. Donald, U.S.N.

Besides the experience gained and the lessons taught by the war, the United States Navy has added to its fleet over two hundred and fifty destroyers in which are incorporated, despite their rapid construction, all that is latest and best in engineering to suit their needs. The building of the battleships, battle cruisers and scout cruisers, planned just before our entrance into the war, has necessarily been delayed, but is now being prosecuted. The machinery of these vessels constitutes the highest development for installations where very large powers are required, and is, therefore, of as much interest today as it was when it was planned. In fact, the planning and improvement of these installations were continued throughout the war and are only now nearing final settlement. The destroyers laid down by this country in such numbers to answer the Hun submarine problem are not yet all delivered, and their engineering characteristics are up to the minute for very light weight, high powered requirements.

Without question, the navies of the world have been the clearing houses for the development or rejection of new ideas in marine machinery, and this is one of the important peace-time functions of the United States Navy. Not only are various inventions and improvements given a trial under conditions where the operators are above the standard of the merchant operating personnel, and where their true worth is demonstrated beyond a doubt, but also the necessary encouragement is given the machinery manufacturers of the country under conditions simulating Government subsidy. The Government itself does a great deal of experimental and research work which has helped the art of marine engineering. Of course, there are certain conditions for which naval machinery must be designed which are not met with in merchant practice, notably, design to provide for economy at low power (cruising speed), and duplication of units, but, in general, the most up-to-date naval practice may properly be considered to be the future trend of merchant vessel machinery. It is hardly likely that any types of propulsive or auxiliary machinery or boilers will be introduced into the merchant service which have not first been proven to be successful in the navy from the standpoint of both economy and reliability.

THE WAR CREATES A MERCHANT MARINE.

The American merchant marine as it stands today is purely a product of the war. The majority of the vessels were built for the Emergency Fleet Corporation. Due to the necessity for quantity production of cargo carriers on a large scale, the standard ship was evolved. If it had been

possible in this country, as it was in Great Britain, where the shipbuilding industry was so well developed, the regulation merchant vessel equipment of triple expansion, reciprocating engines and Scotch boilers would have been installed in practically all ships. As it was, although all facilities for the manufacture of the regulation engines and boilers were used, and the Government contracted for shops and even whole plants for the manufacture of Scotch boilers, still the demand far exceeded the supply and the old standard installations were supplemented by large and small tube water-tube boilers, geared turbines and internal combustion engines. Thus there were adopted for merchant use machinery and boilers which in the ordinary course of events would not have come into general use for several years.

The merchant sea-going engineer is loath to take up "new-fangled" ideas. It is only natural that he should be more averse to them than his brother ashore, who can obtain the advice of experts or the services of the manufacturer's repair men by sending a telegram or using a telephone. In a ship, when she leaves the dock, the engineer must be sure of his machinery; for he is entirely independent of outside assistance, and, in addition, there are the perils of bad weather should a mishap occur. The wireless telegraph has done a great deal to allay these apprehensions.

The personnel situation in the merchant marine, particularly in regard to engineers, is a very serious one even now. The inability of personnel of little experience to operate the newer types of machinery has caused much destructive criticism where it was not deserved. While the failure of a new type of machinery is spread broadcast, that of the old is not considered to be of sufficient importance or interest to warrant discussion, and it is not generally heard of. But it may be safely stated that there have been as many casualties with the older types of machinery as there have been with the newer ones. One point in regard to inexperienced personnel that is worth bringing out is that this same inexperience and newness to marine engineering will make the operating engineers more receptive of the more advanced ideas than would engineers that had been brought up at sea.

The increasing cost of fuel is compelling shipowners to seek for and to adopt any improvements in machinery installations that will give economy, providing reliability is established. The recent development of marine machinery has been along the lines of securing greater efficiency of the unit and greater economy in overall operation. There are some who will say that a number of the late improvements have still to prove their reliability. Recent development has not been in the nature of making machinery more complicated, but just the reverse, and it can be stated from personal experience that the operation and upkeep of a highly modern installation is a much easier task than that of caring for one of the same power, built along the lines in practice ten years ago.

MACHINERY FOR MAIN PROPULSION.

The various types of machinery in use today for main propulsion are as follows: Reciprocating steam engines, turbines, direct connected; turbines, with mechanical single or double reduction gears connecting them to the propeller shafting; turbines, driving electric generators which in turn supply current for turning motors connected to the propeller shafting, either direct or through mechanical reduction gears; internal combustion engines connected to the propellers direct or through mechanical or electrical reduction.

THE RECIPROCATING STEAM ENGINE.

The reciprocating steam engine is the old standard propulsive machinery for merchant vessels of the cargo type. This engine has been developed to what may be considered its maximum efficiency. For slow ship speeds and low powers it has much to recommend it from its reliability. The usual type is triple expansion, three-cylinder vertical, inverted cylinders. Quadruple expansion is also in use. One of this type is that found in specifications for the latest Shipping Board cargo vessels of 10,000, 12,000 and 15,000 deadweight tons. The engines to be used are quadruple expansion, four-cylinder, using the Yarrow-Schlick-Tweedy system of locating crank angles for balancing. An economical figure of .83 pound of fuel oil per indicated horsepower-hour is claimed. The modern naval reciprocating engine is triple expansion, four-cylinder (two low-pressure cylinders), arranged in the order of forward low pressure, high pressure, intermediate pressure and after low pressure, and fitted with forced lubrication for all bearings including valve gear. In general use today are the Stephenson link gear, piston valves on the high pressure and intermediate pressure cylinders, and balanced double-ported slide valves on the low-pressure cylinders (piston valves in the navy), straight, short ports giving minimum cylinder clearances and the lowest possible steam velocities. The strokes used for slow speed cargo vessels are: 36 to 40 inches for engines of 1,000 to 1,250 indicated horsepower, 40 to 48 inches for engines of 1,300 to 1,800, and 48, 54 to 60 inches for larger sizes. The revolutions per minute with reciprocating steam engines for ship speeds of 9 to 12 knots are 90 for indicated horsepowers up to 3,000 and 80 down to 65 for indicated horsepowers of 3,000 and up.

DEVELOPMENT OF THE STEAM TURBINE.

In 1905 the Navy Department determined to try the turbine for marine propulsion. Three scout cruisers were authorized, one to have reciprocating steam engines on two shafts, one to have Curtis turbines on two shafts, and one to have Parsons turbines on four shafts, all using direct drive of the propeller. On trial of these vessels, the reciprocating engine showed up to better advantage than either turbine. However, the results obtained and subsequent improvements in design warranted further trial of the turbine in other vessels.

During the tests of the next few years it still appeared that the reciprocating engine was better suited than the turbine for marine use. This was because the designers of the turbine lost sight of economy of propulsion and aimed only at securing economy of the turbine. A number of methods of securing economy of the turbine at cruising speed were developed, including extra turbines for use at low power only and a combination of turbine and reciprocating engine. These methods prevailed until the development of reduction gears.

By the introduction of such gears, whereby the turbine speed could be kept high at all speeds of the vessel and the propeller speed kept low, large increases in the economy of propulsion resulted, with the further result that the turbine firmly established itself in favor. This discussion applies to cargo vessels as well, for it must be remembered that the cruising speed mentioned is around 12 knots.

This variation of turbine and propeller speed has been accomplished in two ways—by mechanical reduction using helical gears and by electric reduction having the turbine drive a generator which drives electrically a

motor on the propeller shaft. The first trial of the mechanical reduction gear in the United States Navy occurred in 1910, and the trial of the electric drive was in 1911, the ships used being naval colliers. Both of these trials were highly successful; and although there have been a number of direct-connected turbine installations put into naval vessels since that time, this method of drive is now no longer considered. The few destroyers built during the war at Newport News with direct drive of main turbines and geared cruising units were only allowed because of the difficulty of securing the gears for the main turbines; all of the remainder of the destroyers built during this time have geared main turbines.

REDUCTION GEARS.

In the naval service, it is now standard to use mechanical or electrical reduction on all vessels that are to be operated by a naval crew. Electrical reduction is used only for capital ships, *i. e.*, battleships and battle cruisers. Both the reliability and the economy of the turbine have been established from a naval point of view. Geared turbine drive on destroyers has raised the economy of propulsion at cruising speeds over 30 per cent, and in larger vessels the saving, while not so great, is a very substantial figure.

The 20,000-ton, 28,000 shaft horsepower battleship *North Dakota* is an interesting case where the main turbines have been of both types. The original turbines were direct drive, installed in 1910, and were replaced in 1918 by mechanical-geared turbines, one high pressure and one low pressure on each shaft. It is stated in the last annual report of the chief of the Bureau of Steam Engineering that the gain in economy of propulsion effected has amounted to 31 per cent at practically all speeds.

If it were not for the turbine itself, such high powers as are in use or contemplated on battleships, battle cruisers, scout cruisers and destroyers would be out of the question. For example, the powers to be used and in use in our latest high speed warships are as follows: Destroyers (35 knots), 28,000 shaft horsepower; battleships (23 knots), 60,000 shaft horsepower; scout cruisers (35 knots), 90,000 shaft horsepower; battle cruisers (34 knots), 180,000 shaft horsepower. Truly these are stupendous and almost unbelievable figures, if viewed with the light of five years ago. Speeds as high as 39.7 knots (H.M.S. *Tryon*) have been obtained with turbine installations in destroyers. As high as 25,000 horsepower per shaft, and 15,000 horsepower per pinion are now in use.

EFFICIENCY OF MECHANICAL REDUCTION GEARS.

The efficiency of the mechanical reduction gear has been the subject of much discussion, but, with good design, a figure of 98½ per cent for single reduction and 97 per cent for double reduction gears at full power seems safe. The angle of the teeth has been reduced from 45 degrees to 30 degrees in a number of recent designs and a small pitch appears to be preferable to a large one, provided the tooth pressure is not allowed to go too high. In general, gears have been satisfactory in service, and cases of failure can all be traced to hurried manufacture, defective material or carelessness in operation. Pitting of the teeth, due to excessive local pressure, causing spots of small diameter to flake off, has not affected the working or the life of the gears. The most successful gear manufacturers use horizontal machines for cutting gears and some have an attachment for

causing the table to creep, thereby discounting any small error in cutting which would otherwise produce objectionable vibration and noise when the gear is in use.

Not only has the geared turbine proved to be an unqualified success in naval vessels, but also it has been taken up extensively for the propelling machinery of merchant vessels of both cargo and passenger types. Once its reliability is established beyond a doubt when operated by a merchant crew, it will be the only design considered where steam is to be employed, with the possible exception of turbo-electric drive, and where the prospective operating personnel is of a low order. Of all the vessels operated by the Shipping Board, 40 per cent are geared turbine drive.

TROUBLES WITH GEARED TURBINES.

It is reported that a great deal of trouble has been experienced with the gears of these vessels, and it is because of this that the Shipping Board in its latest vessels is considering only the quadruple expansion, reciprocating steam engine. In all fairness to the gear manufacturers, it must be stated that these troubles were due either to the hurried manufacture of the gears under stress of war conditions, demanding the full capacity possible of the shops, or were due to the inexperience of the operating engineers, who probably would have come to grief with reciprocating engines.

LUBRICATING SYSTEMS.

A material aid to the successful operation of the geared turbines on merchant ships is the new arrangement of the lubricating system being introduced into these vessels, and troubles undoubtedly will be less frequent when it is adopted on all geared turbine ships operated by merchant crews. Lubrication of the main propelling plant is a very important item in any system, but it is of paramount importance in turbine drive, whether direct or geared. The choice of lubricating oil is well handled by the large lubricating oil companies, and, for the merchant service a ship is hardly likely to get the wrong kind of oil for the type of engine installed.

The system of lubrication originally installed on merchant ships has not proved altogether satisfactory, and the failure of many geared turbine installations has been due to this fact. In the naval service, it is standard practice to supply the lubricating oil by duplicate pumps under pressure direct to the bearings and gears. This method has proved a most satisfactory one, and there are few cases of failure of the system. In the merchant service it has been found necessary to install tanks of a considerable capacity at a height above the engines that will insure a proper pressure on the system to the bearings, 20 feet, or about 10 pounds, being considered a minimum figure. The tanks are fed by the forced lubricating oil pumps and an alarm (float type, electric) is fitted on the tank to give due warning when the supply is getting low.

For merchant practice, two methods of installing the pumps have been used; one, with the pumps driven from the main shaft, and, the other, with pumps having independent drive, which are provided in duplicate. Strainers in the oil discharge piping are always installed, and, in naval practice, in the suction piping also. The application of the centrifugal separator, first used for the separation of cream from milk, for the renovation of lubricating oil marks, really, a great advance in lubrication and eliminates many of the troubles that appear only in marine installations.

The separator removes from the oil both water and sediment in a most efficient and expeditious manner, even when salt water has formed an emulsion with the oil, and delivers the oil in the same state as when new. These separators are being installed by the navy wherever forced lubrication is used. When used properly and regularly, they safeguard the bearings by making it possible to keep the oil continuously in good condition, and there is no reason why, with due care, the bearings should show any appreciable wear even after years of service.

The money value in oil saved has made their installation in naval vessels well worth while. The method in general use of settling out sediment and water by heating the oil in settling tanks is not nearly so satisfactory in that the separation of the impurities from the oil is not complete, the time for settling with any results at all is comparatively very long, and the tanks required are cumbersome.

The single reduction gear is in use where the ship speed is above about 14 knots. For speeds below that, the double-reduction gear is usually found. The average cargo ship recently built has a propeller speed of 90 revolutions per minute for geared-turbine drive.

BALANCE OF TURBINES, GEARS, AND PROPELLERS.

When using high-turbine speeds, as in reduction geared installations, the matter of securing a good degree of both static and dynamic balance is of much importance. This matter has been taken up by the navy at a recent date, and although only a few installations have been scientifically balanced, the necessity for it is recognized, and it will become standard practice. The most successful balancing has been done by the Mare Island and New York Navy Yards, where not only are the turbine rotors balanced, but also the pinions, main gears, shafting and propellers. The perfection of balance obtained has been such that when the turbines are driven at 20 per cent over speed (with propeller shafts disconnected), the vibration is so small as not to shake down coins standing on edge on the bearing-cap bolts. Destroyer geared turbines which have been balanced show an increase in efficiency over those which are not, while the vibration of the hulls becomes reduced to a negligible minimum. Proper balancing also has a marked effect on reducing repairs to the turbines.

THRUST BEARINGS.

In connection with the geared turbine, due credit must be given to the Kingsbury thrust bearing in this country, and to the Michell thrust in Great Britain. This type of bearing makes possible the use of a single collar. It gives a decided improvement in the mechanical efficiency, saves space and weight and gives economy of lubricating oil. It is suitable for use with any type of propulsion. While it has not by any means come into general use in the merchant service of this country, there are a few vessels in which it has been installed. The reason for it not being adopted on a large scale is the same reason that holds back the geared turbine, *i. e.*, lubrication—oil must be supplied continuously. In the naval service, it has become the standard thrust bearing with the building of the *Tennessee* class of battleships. All destroyers built during the war are fitted with it, and, in fact, had it not been available, serious difficulty would have been met with in finding sufficient space for the old horseshoe type. The General Electric turbines supplied to the navy have used their type of single-collar roller thrust bearing on the turbines themselves.

The bearing pressures used in the navy for the horse-shoe thrust were from 69 to 74 pounds per square inch, whereas, with the Kingsbury, the thrust pressures run from 291.5 to 425 pounds per square inch. The 425 figure is found on the new turbine installation of the *North Dakota*, where these bearings have been particularly successful.

Hydraulic reduction gear is in use on some German ships (Föttinger type), and has also been tried experimentally in the British Navy. It is not used in this country in either the merchant marine or navy.

TYPES OF TURBINES.

The two general types of turbines in use are the Parsons and the Curtis, with modifications of both. In America, the Curtis is found in more installations than is the Parsons, both in the navy and in the merchant service. However, the Parsons installations in the navy have all been most successful, and the more general use of the Curtis is rather a question of its having been developed on a larger scale by the manufacturers in this country than has the Parsons. The design of blading and particularly the method of securing blades in the rotor have been much improved. In England it is usual to find the Parsons turbines that have recently been built fitted with an impulse type of first stage, in order to reduce complications due to very short first-stage blades, and consequent high percentage of leakage past the blades.

ELECTRIC DRIVE.

Another method of obtaining both high propulsive efficiency and high turbine efficiency at the same time is the reduction by means of electricity of the speed of the propeller below that of the turbine shaft. This system of propulsion offers such great advantages in the matter of location of the machinery to obtain maximum protection from damage due to under-water attack that it has recently been adopted by the navy as the standard for capital ships, *i. e.*, battleships and battle cruisers. The engineering features have been well developed by both the General Electric and the Westinghouse companies, and this country is much farther advanced than any other in knowledge of the details of design and manufacture connected with such installations.

So far, the only cargo vessel in America on which it has been installed is the *Jupiter*. This vessel was the one selected by the navy for the demonstration of electric propulsion, and the result of about six years' service has demonstrated the reliability of the system beyond a doubt. It was after the successful test of the *Jupiter* that the design was adopted for the *New Mexico* and later for twelve battleships and six battle cruisers now building. At present the navy does not contemplate the installation of electric drive on any vessels except capital ships.

The Coast Guard Service is placing this drive in two of the five 1,600-ton, 2,600 shaft horsepower cutters now building, the other three ships to have mechanical-geared turbine drive.

In England, electric drive has not received much attention, although one merchant vessel is so fitted. On the British *K*-class submarines, it has been adopted for use immediately after coming to the surface or just before submerging, the motors being driven at this time by Diesel engines. When operating normally on the surface steam turbines are used and when submerged, of course, the motors are operated by batteries. In Sweden, seven vessels are being fitted with Lungström turbo-electric drive; two are of 1,700 horsepower and five of 1,200 horsepower.

It is understood that the Shipping Board is considering installing electric drive on some of its cargo vessels. For merchant service, the Diesel electric drive is looked upon by some with more favor than the turbo-electric combination. An interesting recent installation of this character is on a trawler just put in service on the New England coast.

ADVANTAGES OF ELECTRIC DRIVE.

The merits of the electric-drive system as applied to naval capital ships are well stated by Admiral C. W. Dyson, chief of the design division of the Bureau of Steam Engineering, Navy Department, in the May, 1917, issue of the JOURNAL OF AMERICAN SOCIETY OF NAVAL ENGINEERS. It will be seen that many of the advantages stated apply as well to merchant vessels.

1. Greatly increased torpedo protection for ships.
2. Greater flexibility in machinery arrangement.
3. Better and wider separation of important units.
4. Minimum lengths and diameters of steam pipes.
5. Reduced heating of vessel from steam pipes.
6. Better centralization of power.
7. Fewer bulkheads pierced by steam and feed piping.
8. Reduced engine room complement.
9. Elimination of danger from fractures of piping due to shells striking protective deck.
10. Greater ease in control.
11. Greater flexibility in power distribution.
12. Better maintenance of economy through a wide range of powers.
13. No metallic contact between rotor and stator of motor.
14. Eliminates all dangers of disarrangement due to shaft vibration, when the helm is put hard over.
15. Maximum reduction in length of shafting.
16. Increased backing power.

The internal combustion engine for ship propulsion has not received the stimulus that it deserves in this country. The Shipping Board has two vessels in service equipped with Diesels, one a 4,500 indicated horsepower Burmeister and Wain and the other, a 3,000 indicated horsepower McIntosh and Seymour installation. It has been proposed by the Shipping Board to install this type of engine in ten other vessels.

MOTORSHIPS.

Very few vessels are equipped with internal combustion engines of over 1,000 horsepower per set. However, that there is some and increasing interest taken in this country in the Diesel engine for marine propulsion is shown by the fact that most of the large shipbuilding companies are obtaining licenses to manufacture foreign designs and a few are developing engines of their own design. The war caused a setback in the development of the Diesel for commercial work, which will make it more difficult for it to catch up with the geared turbine. Considerable experience was, however, gained during the war in the building of heavy oil engines for submarine work, and undoubtedly the high cost of fuel will force it to the front in considering types of propulsion of future cargo ships.

PROPELLERS.

The form of the propeller in use today is the same that has been in use for years, that is, the true screw is still considered to be the proper form

and the many inventions and attempts to improve on it have come to nothing. However, the method of preliminary determination of pitch, diameter and projected area has been so improved that predicted performance checks up very accurately with actual results where the form of hull is known in advance.

BOILERS.

The main features in boiler development for marine use have been the introduction of the water-tube boiler into merchant ship work, the great extension in the use of fuel oil in place of coal and the development of the small-tube boiler of very high capacity for naval purposes.

Upon the entrance of the United States into the recent war, when the construction of merchant vessels was taken up by the Government on such a large scale, it was impossible to secure a sufficient number of the standard marine Scotch (or firetube) boilers, and it became necessary to call on manufacturers of water-tube boilers to outfit many of the new vessels. As a result, of the total number of boilers in the Emergency Fleet about 56 per cent are water-tube and about 44 per cent are fire-tube. The water-tube boiler is forcing recognition as the superior type for merchant work, and one hears more and more of its merits. This is especially true where oil fuel is used and where pressures over 200 pounds gage are necessary. However, the specifications for new cargo vessels recently issued by the Shipping Board call for the boilers to be single-ended Scotch, steam pressure, 220 pounds gage, 100 degrees F. superheat and oil-fired.

A large number of types of water-tube boilers are found in recently constructed merchant vessels, and the type in most general use is the Babcock and Wilcox, with the Emergency Fleet boiler a close second. The customary evaporating rate used in merchant work is 5 pounds steam per square foot heating surface for both water-tube and fire-tube boilers, whether oil- or coal-fired.

Mechanical stoker firing of water-tube boilers has received practically no encouragement, although there is no apparent reason why some of the designs which have met with such success ashore should not be developed for marine purposes. The navy has not attempted to develop it because of failure of existing designs to stand up when the vessel is rolling, because of the sudden and frequent variations in load required when ships are steaming in formation, and because of the adoption of oil as fuel. The use of mechanical stokers on merchant ships will probably come when the price of coal drops, for the advantages of mechanical stoker firing are many where the load is constant hour after hour, and where the firemen are more apt to be inefficient.

NAVAL BOILERS.

In the naval service the large tube water-tube boiler is the standard for all installations except where, on account of limited space or weight requirements, the use of the small-tube express-type boiler is mandatory, and in auxiliary vessels which have merchant crews where Scotch boilers are still installed in a few cases.

Upon the adoption of oil fuel for naval boilers, the Babcock and Wilcox Company, in order to meet the demands of the navy, modified their boilers as follows: The side-water walls were eliminated, heavy fire brick walls being substituted; the angles of the tubes and headers were changed, tubes from 15 degrees to 18 and 20 degrees and headers accordingly; outside downcomers were installed between the steam drum and the lower

front crossbox; the lower row of 4-inch tubes was done away with and two rows of 2-inch tubes substituted. At a very recent date it was found necessary to modify the baffle in the steam drum in order to prevent priming at very high rates of steaming for this type of boiler.

In addition to the new express boiler designed by Admiral Dyson, which is made mention of later, improvements have been made in an express boiler of the White-Forster type, fitted with superheater. On test at the fuel-oil testing plant at Philadelphia in May, 1919, this boiler (total heating surface 8,353 square feet) obtained an evaporation of approximately 23 pounds of water from and at 212 degrees F. per square foot of heating surface with an equivalent evaporation of 15.3 pounds of water per pound of fuel oil. This is a record high capacity performance. It is of much value in that it proves that the express-type boiler may be forced, without danger, to a much higher capacity than was formerly thought possible. In the battle cruisers, scout cruisers and latest type of battleship, where very high powers are used, adherence to former practice would have made the boiler weights prohibitive.

FORCED DRAFT.

Forced draft is becoming more general in merchant ship boiler installations, whether using oil or coal fuel. The air pressure is 2 inches of water regardless of the fuel. For coal, the Howden closed-ash-pit system is used and for oil the air is supplied by ducts led to each individual burner. In the naval service, a pressure of 2 inches of water is used for coal and 6 to 9 inches for oil. Eight inches has been recently adopted as the standard pressure for capital ships in order to prevent flare back of oil flame into the fire room due to the blast down the smoke pipe from heavy gunfire. On the new 35-knot destroyers, one often sees as much as 9 inches air pressure in the fire rooms when the boilers are driven at full power. Naval practice has always been to use the closed-fire-room system of forced draft, and this will undoubtedly be adhered to in the future on account of safety and comfort, although the air leakage is in design estimated at 50 per cent notwithstanding the great care in construction given to secure air tightness of compartments. In the latest capital ships, this figure is probably too high, for each boiler is in an individual compartment and the only access is through a well-constructed air lock.

Steam pressures have not been materially increased. In the naval service the tendency has been, if anything, toward a slight reduction in pressure. In the merchant service it is, however, now common to use pressures of from 200 to 250 pounds gage, about 220 pounds being the usual figure. Of course, when water-tube boilers are installed the pressure is apt to be higher than when firetube boilers are called for.

SUPERHEATED STEAM.

The use of superheat is becoming recognized to be well worth while from the material increase in economy that is realized, and an increase in the amount of heat per pound of steam available for useful work is being obtained by superheat rather than by increase of pressure. In the navy, superheat was introduced in recent years in the *Michigan* class of battleships and continued into the *Delaware* class only to be dropped, on account of the use of Parsons turbines, and then to be taken up again in the new battleships and battle cruisers now building.

In the battle cruisers particularly, where it was necessary to install an express-type (small tube, large angle of inclination to the horizontal) boiler, and as there was only one boiler of this character on the market to which superheaters were adaptable, the Bureau of Steam Engineering designed a boiler especially for the purpose. This boiler was brought out by Admiral Dyson, and the particular feature of design is that the superheater is nested in the generating tubes, but in such a manner that it can be rolled out for the renewal or inspection of tubes.

Superheats of 100 degrees F. are common in this country. In England the tendency is to use 200 degrees. All of the later Shipping Board merchant ships are fitted with superheaters. This applies for both turbine and reciprocating engines, water-tube and fire-tube boilers. The Shipping Board prefers the waste-gas type superheater for water-tube boilers, while the navy installs the superheater at the end of the first gas passage in a baffled water-tube boiler. In fire-tube boilers the most favored superheater consists of tubes laid within the upper rows of generating tubes.

FUEL OIL BURNING.

Fuel oil may now be taken as being the standard fuel for the navy for all types of vessels. In the merchant service fuel oil is most popular, as is indicated by a survey of the ships operated by the Shipping Board, where of those fitted with reciprocating engines 70 per cent burn fuel oil and of the turbine ships 55 per cent use this fuel. Some interesting figures in this connection are the standard fuel allowances per horsepower-hour for the various types of ships in the Emergency Fleet. These figures are used for checking fuel requisitions and the performance of vessels.

Type Engine.	*Fuel Used*	*Standard Fuel Allowance in Pounds Horsepower Hour*
Reciprocating	Coal	2.41
	Oil	1.25
Turbine, geared	Coal	2.0
	Oil	1.0
Diesels	Oil*	.39

Important points in the development of fuel oil burning in the navy are as follows: The superiority of the Bureau of Steam Engineering and the Peabody types of registers and atomizers has been demonstrated beyond a doubt by test and service use. The bureau register may be used for either forced or natural draft.

A large variety of types of registers and atomizers are in use in American merchant ships. The Shipping Board vessels alone have fifteen types installed and the tendency there is to use natural draft burners. The new naval general machinery specifications provide that the center of the register must be 22 inches from tubes or sides or bottom of furnace. Specifications for latest naval capital ships make provision for a maximum of 10.8 pounds of oil to be burned per cubic foot of furnace volume. The matter of having a sufficiency of furnace volume has been found to be one of utmost importance if good combustion is to be obtained.

AUXILIARY MACHINERY.

Improvements in the design of auxiliary machinery have kept pace with the advance of main propulsive machinery.

* Includes 150-horsepower boiler in use for steam heating.

CONDENSERS.

The standard style of condenser that has been in use for years is still the rule. However, a number of modifications have recently come into use. The cross section is either circular or pear-shaped. Merchant practice uses straight ¾-inch tubes packed at both ends, naval practice uses straight ⅝-inch tubes expanded into the tube sheet at the entering-water end and packed at the other, or curved ⅝-inch tubes (destroyers) expanded at both ends. Two passes of the circulating water are the rule in merchant and naval ships except destroyers and scouts, which use one pass. In merchant practice the water enters at the top in some and at the bottom in others. Latest merchant ships are provided with two injections, one high for use in shoal, smooth water, and one low for use at sea. On all the lake steamers that were brought to the seaboard for use in trans-oceanic service it was necessary to fit a low injection as the one installed rolled out of water when the ship was in a seaway.

Naval two-pass condensers recently constructed have spherical heads. It was found that the flat head gave too little area for change of direction of flow of water at the reversing end. In the latest naval condensers the cooling surface per horsepower has been decreased and circulating water considerably increased.

The only departure worthy of mention from the standard arrangement of tubes and steam flow is the Lovekin patent condenser. This is a special design with tubes and steam spaces baffling so arranged as to provide, by use of drip sheds, for a minimum loss of effective cooling surface by drowning of the lower tubes from condensed steam dripping from the upper tubes and to provide for a central vertical chamber in the tube nest from which the air ejector takes suction, the chamber being open to radiating passages which draw from all parts of the lower half of the condenser. This type of condenser is being installed in the battleships *Tennessee*, *California*, *Colorado* and *Washington*. It is also to be used in twelve army transports building at Hog Island. One of the latter vessels is now testing the condenser. It is expected that the vacuum desired will be obtained with a considerable reduction in cooling surface and circulating water over the old type without special baffling in the steam space. Also it is expected that the vacuum obtained at the top of the condenser will more nearly equal that at the air ejector suction.

PUMPS.

For air pumps the Edwards type is still the favorite in the merchant service for both turbines and reciprocating engines and for the latter is operated by the engine. There are, however, a few installations of the air ejector—condensate pump type for turbine installations. This latter system is the only one considered for the latest capital ships of the navy, both on account of the great saving in space and weight and also on account of the ease with which a high vacuum may be maintained.

The augmenter type of ejector discharging into a special condenser connected to the suction of a reciprocating air pump has been, in some cases, displaced by the two-stage ejector, discharging to an air separating space in or above the main feed and filter tank. In this type, all heat in the steam used for operating the ejector may be reclaimed in the feed water. Inter-condensers are sometimes installed between the first and second

stages of the ejector in order to reduce the steam consumption, but they are not necessary if the heat in the steam used does not raise the temperature of the feed water to too high a degree.

FEED PUMPS.

Merchant ships are using vertical, simplex, double-acting type feed pumps, independent of the main engine. In naval practice the same type is used for auxiliary feed purposes, but for the main feed turbine driven centrifugal pumps are being installed on large vessels.

In naval service the centrifugal pump, either steam or motor driven, is coming into use wherever this type of pump is practical. The Bureau of Steam Engineering, Navy Department, has given a great deal of careful study to the question of speed control for turbine-driven auxiliaries supplemented by actual tests and experiments on board vessels of the fleet, with the result that satisfactory governing devices have been selected.

In the merchant service, pumps, except main air, feed and circulating, are usually of the horizontal, duplex, double-acting type, but even here the centrifugal turbine or motor-driven pump is coming into use and is found in not a few new turbine-driven ships.

On motorships, motor-driven pumps, and in fact motor-driven auxiliaries of all types are the rule.

FORCED DRAFT BLOWERS.

The turbine-driven blower is the standard for naval vessels when forced draft is installed. This has replaced the motor-driven blower principally on account of its greater range of and better elasticity of speed control, and the fact that it is independent of the ship's generators.

Merchant ship blowers are usually driven by reciprocating engines; there is a tendency, however, to install motor drive.

FEED WATER HEATERS.

Feed-water heaters are now always installed on naval vessels. The type is the closed heater on the discharge side of feed pumps. Straight, "U," and coiled-tube types are used. The specifications have recently been changed to provide for a capacity 25 per cent greater than is necessary to obtain the required feed temperature at full power and drain of such size as to handle twice the condensed steam expected. For feed-heating reasons, the pressure on the auxiliary exhaust system is carried at a standard of 10 pounds gage, and steam not condensed by the feed heaters is spilled into the main engines, with an emergency relief to condenser.

Feed heaters are being installed in the most up-to-date merchant vessels, the installation being similar to naval practice except that the pressure carried on the auxiliary exhaust system is 5 instead of 10 pounds gage.

EVAPORATING (DISTILLING) PLANTS.

These have recently received a great deal of attention and improvement in the navy. The low capacity (25 gallons per day per square foot heating surface), horizontal straight-tube type has been abandoned for the high capacity per square foot heating surface, vertical coiled or "U" tube types. The latter type of evaporator has been developed to a remarkable

degree and capacities of 120 gallons per square foot heating surface per day have been made. Naval practice now limits this to 60 gallons per day per square foot heating surface, except on destroyers, where as high a figure as 80 is allowed.

On naval vessels it is usual to install evaporators for operating on live steam using the double-effect system. Latest installations allow for the use of auxiliary exhaust in coils, the evaporators being operated single effect and an air pump being provided for maintaining about 20 inches of vacuum on the distilling condensers. This has proved to be an installation producing great economy.

LOW PRESSURE DISTILLING PLANTS.

It is not customary to use more than double-effect operation, but an exception to this is the interesting installation in the battleships *Tennessee* and *California*, to be commissioned in the near future. With the exception of the installation in the *Dixie*, these are the only all low pressure distilling plants in the navy. Six evaporators are installed and piped up so that they may be operated in single, double, triple, quadruple or sextuple effect. Live steam may be admitted to first effects and auxiliary exhaust may be admitted to first effects for single effect operation or to the second effect in multiple effect operation.

The daily capacity of naval distilling plants has been materially increased, destroyers from 5,400 gallons to 10,500 gallons in the latest boats and on battleships from 25,000 gallons on the *Pennsylvania* class to 50,000 gallons on the 49-54 class (60,000 shaft horsepower), and 70,000 gallons on bottle cruisers 1 to 6 (180,000 shaft horsepower). An allowance of 10 gallons per man per day is considered with the navy a liberal allowance for all purposes except for engineering department, and in the engineering department on large vessels an approximate figure for making up feed expenditure is 1,800 at anchor, and 4,200 under way at 12 knots (varying from this with the power required). These are not design figures.

Practically all merchant ships recently built have evaporating plants of considerable capacity installed. The capacity is now such that the vessel may, if desired, distill all of the water for both deck and engineering purposes.

REFRIGERATING PLANT.

On merchant ships built during the past few years, a refrigerating plant of ample capacity is always found in almost all cases. On cargo ships this is about two tons per 24 hours. Ammonia compression, ethyl chloride, CO_2 and SO_2 systems are the rule, with preference for SO_2 in the Shipping Board vessels. Steam drive is being displaced by motor drive. The dense air system has until recently held the field in the navy. CO_2 systems are now installed on practically all new ships, except destroyers, where ethyl chloride and SO_2 systems are found with a few of the CO_2 type. Naval refrigerating apparatus of whatever type is turning to motor drive, especially for small-size systems of two tons and under.

ELECTRICAL PLANT.

Aboard ship electricity is being put to more and more purposes and the electrical plant is a most important part of the installation. The geared turbo-driven generator is the naval practice, while the reciprocating steam

engine-driven generator is found in most common use on steam merchant vessels, with turbo drive on a few. Ships with internal combustion engines for main propulsion commonly use the same type of engine for generators.

The sets are now installed in duplicate on merchant ships, this having always been the naval practice. It is believed that the introduction by the navy of interior communication on commercial vessels during the war will have a lasting effect in that the need of the telephone, call bell, and electrical indicator for various purposes will be felt by merchant ship captains. The electric motor-driven pump which is being taken up on these vessels will also mean an extension of the electrical plant. Radio installations on merchant ships are now looked upon as a necessity.

CONCLUSION.

In summing up the situation, one need not be a born optimist to feel that tremendous strides have been made in the general improvement of engineering at sea, and that future developments as indicated by conditions today will be most interesting.—"International Marine Engineering."

TWO YEARS OF ELECTRIC PROPULSION ON THE *NEW MEXICO*.

As the *New Mexico*, the first United States battleship to be fitted with electric drive, was placed in commission at the New York Navy Yard in May, 1918, she is now just completing her second year of active service. During that time she has seen service of practically every kind that is encountered by a battleship except actual engagement in battle. The results of this service, according to Commander S. M. Robinson, U.S.N., fleet engineer of the Pacific Fleet, of which the *New Mexico* is the flagship, have been highly satisfactory and justify the judgment of those who are responsible for the installation of electric machinery in the vessel.

As Commander Robinson gave a very complete description of the *New Mexico's* machinery in this magazine a year ago, it will be remembered that the electric machinery for propulsion consists of two main turbo generators rated at 11,500 kilowatts each at 80 per cent power factor and having an overload capacity of 25 per cent, four main motors rated at 7,250 horsepower each and having an overload capacity of 25 per cent, two boosters for varying the current of the main field, two 300-kilowatt exciters for supplying the current to the main field and also certain electrically-driven auxiliaries, together with a main switchboard, an exciter switchboard and the necessary wire and cable. The ship uses one generator for speeds up to 17 knots and two generators for speeds from 17 knots to full speed, which is between 21 and 22 knots. For speeds up to 15 knots the motors are run on the 36-pole connection, so that at 15 knots the turbine is running at its designed full speed; from 15 knots to full speed the motors are run on the 24-pole connection. This arrangement gives good economy at all speeds, and in order to give our readers a definite idea of just what results have been obtained, Commander Robinson has prepared the following report:

The *New Mexico* has been operating for nearly a year in company with two sister ships, the *Idaho* and *Mississippi*, which have hulls identical with

that of the *New Mexico*. During this time it has been possible to get an accurate comparison of the relative economy of the three ships and also the relative maneuvering qualities. In the latter respect, the *New Mexico* is decidedly superior, and the remarkable part of it is that nearly all the maneuvering in restricted waters has been done with one turbo-generator. When this installation was first proposed, its opponents maintained that, while a ship like the *Jupiter* could be satisfactorily operated with the screws on both sides of the ship running at exactly the same speed, it would not be possible to get satisfactory operation with that arrangement on a ship which had to operate in formation. But exactly the reverse has proved to be true; it has been found that more satisfactory operation is obtained when using one generator than when using two, and it is customary, when in dangerous waters where it is desired to take all possible precautions, to use one generator for driving the ship and to keep the other turning over idle. If the ship is getting under way from an anchorage and has to turn, as soon as the anchor is away the signal is given for standard speed ahead on one side and the same speed astern on the other; with this arrangement the ship will turn absolutely on her heel without gaining ground either ahead or astern; with other engines, where it is not possible to regulate the speed so quickly and accurately, the probability of getting speed on the ship in one direction or the other is much greater. All the predictions in regard to trouble on account of the condition that when operating with one generator the screws on both sides of the ship must run at the same speed (if run at all) have proven to be groundless.

In regard to the comparison of economics the results have been very favorable to the *New Mexico*. The *Idaho* and *Mississippi* are fitted with direct-connected turbines (one is Parsons and the other Curtis) and have geared cruising turbines; the *Idaho* can use her cruising turbines at all speeds up to about 17 knots and the *Mississippi* can make about 15 knots with her cruising turbines. Experience gained by a comparison of destroyers fitted with geared main turbines and those fitted with direct-connected main turbines and geared cruising turbines indicates that the economy of the *Idaho* and *Mississippi* at cruising speeds is about as good as it would be if they had geared main turbines. A comparison of the *New Mexico* with these ships is therefore particularly interesting at the lower speeds.

The advocates of electric propulsion have always claimed that it was very superior to all other forms of propulsion at the cruising speeds, but even the most enthusiastic of these have been surprised by the remarkable showing made. This is doubtless due to the fact that no one made sufficient allowance for the saving due to shutting down one generator and all the auxiliaries that go with one of the condensing plants. At a speed of 10 knots the *New Mexico* uses about 16.7 per cent less oil than her sister ships, or, putting it another way, her sister ships use about 20 per cent more than the *New Mexico*; at 13 knots the figures are 29.9 per cent or 42.7 per cent; at 16 knots the figures are 32.3 per cent or 47.8 per cent; at 19 knots the figures are 28.6 per cent or 40.1 per cent; at full power the figures are 24.4 per cent or 32.2 per cent. At 19 knots and also at full power the *New Mexico* uses about .975 pound of oil per shaft horsepower per hour, and at 15 knots she only uses 1.1 pounds of oil per shaft horsepower per hour. This is a remarkably uniform economy.

The *New Mexico* has just completed her annual full-power trials and the following table gives an analysis of the data obtained:

	Full Power	Endurance
Revolutions per minute	167.4	151.7
Speed (knots)	21	19.35
Shaft horsepower	28,820	21,650
Pounds of oil per shaft horsepower per hour	.975	.973
Pounds of oil per square foot of heating surface	.506	.3795
Estimated water evaporated in pounds per hour	393,500	294,700
Estimated water per shaft horsepower in pounds per hour (all purposes)	13.65	13.62
Pressure (gage) in pounds per square inch at the boilers	270	265
Pressure (gage) in pounds per square inch at the turbines	255	260
Superheat (in degrees Fahrenheit)	32	23
Vacuum	28.7	29.0

In regard to the reliability of the machinery, the *New Mexico* has had nothing but the most minor troubles with her electric plant and there have been no navy yard repairs whatever; there has been one serious accident to the steam part of the machinery which required work to be done by a navy yard, but that was due entirely to a mechanical defect. Due to improper construction of the main governor, one of the weights became detached while the turbo-generator was running without load; the turbine ran away and operated the emergency governor, which tripped the throttle, but the latter did not entirely close and the turbine ran at over-speed sufficiently to stretch the turbine wheels and the entire rotor had to be replaced. The governor has been changed and additional over-speed protection given by arranging for the emergency governor to close the admission valves as well as the main throttle. The machinery has been in operation nearly a year since repairs were completed and no further trouble has developed.

In conclusion, it may be said that the performance of the *New Mexico* since commissioning has been entirely satisfactory in every way and that the expectations of those who were responsible for its installation have been more than realized.—"International Marine Engineering."

NATIONAL MARINE WEEK AND NATIONAL MARINE EXPOSITION.

As predicted in the last number of this JOURNAL, visitors to the National Marine Exposition, held in the Grand Central Palace in New York City, April 12 to 17, 1920, gained a wide view of the American Merchant Marine, the manufacturers interested in it, the shipbuilding yards and the shipping lines.

The Exposition was intensely interesting, and the display included practically every piece of machinery and equipment found on board vessels, together with models of shipbuilding yards and working models of dry docks. That people are interested in the Merchant Marine, was proved by the attendance at the Exposition, which became so great at times, that the doors had to be closed in order to comply with the Police and Fire Regulations.

If the overwhelming success of the Exposition can be taken as a measure of the success of the National Marine week, and it is believed that it can be so taken, the National Marine League has certainly succeeded in arousing the interest of the citizens of this country in the work and needs of the American Merchant Marine, and much credit is due to the League for the success of the celebration.

OBITUARY.

EDMUND MILLS.

The Babcock & Wilcox Company suffered a serious loss of one of its oldest and most faithful employees in the death of Mr. Edmund Mills on January 26, 1920. He had undergone an operation about a week before his death which it was hoped would give relief, but the case was one of serious complications from the start, without much hope.

Mr. Mills was born in Jersey City, N. J., February 15, 1864, so that he was almost fifty-six years of age. Nearly twenty-five years of his life had been spent in the service of The Babcock & Wilcox Company.

He received his education in the public schools of Jersey City, and having great natural mechanical aptitude, after leaving school served an apprenticeship as a machinist and worked for a short time at that trade. Later on he was in business for himself for nearly four years. He was then for about two years in the Drafting Department of the Daft Electric Company, about a year of the time as chief draftsman. From 1891 to 1895 he was with the Colwell Iron Works, which later became the Wheeler Condenser and Engineering Company.

In 1895 Mr. Mills entered the Marine Department of The Babcock & Wilcox Company as a draftsman and was for a time the only one. He had the privilege of close association with Mr. Hoxie in the development of the Babcock & Wilcox Marine Boiler as it now exists by working out the detail drawings to embody Mr. Hoxie's ideas. When the work of the Marine Department grew enough to need an enlarged drafting department he was made Chief Draftsman of the Marine Department, a position which he filled most efficiently and where his splendid mechanical judgment had an excellent field.

In 1909 he was transferred to the Marine Department in the general office as an engineer and correspondent, and here the last eleven years of his life were spent in most efficient and loyal work for the Company's interests.

He not only had fine mechanical judgment, but he had equipped himself thoroughly as an engineer dealing with combustion and the generation of steam, so that he was well equipped for the work which devolved upon him. He was conscientious and careful in the highest degree so that it was always safe to feel that when he had completed the work on any problem it was well done. His intimate service during almost the entire life of the Marine Department had given him a familiarity with all the marine work done by the Company which those coming later on can never hope to equal. His memory was remarkable and until the enormous output during the World War it was safe to say that he knew the number of every marine job and a good deal of its history during the life of the vessel.

Besides his life work as an engineer Mr. Mills had interests for his leisure hours outside of business in which the same painstaking attention to detail gave him rather unusual distinction. His chief avocation was the study of polar exploration, particularly in the Arctic. He had been collecting books on this subject for years and it is believed that his collection is probably the finest and most complete of any private individual. He enjoyed the acquaintance of the late Admiral Melville, and on one occasion showed the Admiral the catalogue of his Arctic library. After looking it over the Admiral told him that included in it were a number of books of which he had heard but which he had never seen himself. We understand that it had been his intention to present this collection to one of the geographic societies or to some public library, and in all probability his family will carry out this intention.

He was also interested in astronomy as an amateur and had built a small observatory at his home where with a telescope of moderate aperture he carried on searches for double

stars and other astronomical pursuits in which an amateur with a small equipment can be of service.

Mr. Mills was a fine specimen of a Christian gentleman of the highest type and was a faithful friend and an agreeable companion to all his associates. He had been an officer and active worker for many years in one of the leading churches of Jersey City and always stood for what was finest and best in American manhood. He leaves behind him an enviable record as a loyal, efficient worker for the Company and as an upright, agreeable man whose memory will always be cherished by those who were close to him as one of the agreeable features of their lives.

He leaves a widow and son to whom the sympathies of all his many friends go out very sincerely.

Mr. Mills joined the American Society of Naval Engineers as an Associate Member in 1902, and continued in such membership until the date of his death.

BOOK NOTICE.

STERLING'S MARINE CATALOG & SERVICE, by FRANK W. STERLING and OSCAR F. COOPER. Size, 9 × 12 inches. Cloth bound, gold stamped. STERLING-COOPER COMPANY, 141 Broadway, New York City. Price, $25.00.

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The following members and associates have joined the Society since the publication of the last JOURNAL:

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Bruner, R. W., Lieutenant, U. S. N.
Munson, R. M., Lieutenant (CC), U. S. N.
Rowe, Gordon, Lieutenant, U. S. N.

ASSOCIATES.

Day, Percy C., 2424 Cedar Street, Milwaukee, Wis.
Gregor, Henrik; Hooven, Owen, Rentschler Co., Hamilton, Ohio.
Ryan, Alexander, 109 Douglas Street, San Francisco, Calif.
Thau, W. E., 567 East End Avenue, Pittsburgh, Pa.
Voorheis, J. T., 263 South Tenth Street, Philadelphia, Pa.
Wilkins, W. N., 165 Eleventh Street, Hoboken, N. J.
Worden, E. P., Submarine Boat Corp., P. O. Box 456, Newark, N. J.

The Board has audited the books of the American Society of Naval Engineers, for the year 1919, and finds that the statement published in the February issue of the JOURNAL is correct.

S. C. HOOPER, Comdr., U. S. N.
K. WHITING, Comdr., U. S. N.
(*)

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* Captain of Engineers W. M. Prail, U. S. C. G., absent from city at time of concluding audit report.

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